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A REPORT ON THE A-SECTOR TEST CRYOGENIC EXPERIENCE OF THE FERMI NATIONAL ACCELERATOR LABORATORY SUPERCONDUCTING ACCELERATOR

AUGUST 1982

FERMI NATIONAL ACCELERATOR LABORATORY
BATAVIA, ILLINOIS

OPERATED BY UNIVERSITIES RESEARCH ASSOCIATION
FOR THE UNITED STATES DEPARTMENT OF ENERGY

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FOREWORD

This report represents the coordinated effort of many Fermilab employees and the first test of a significant part of a superconducting accelerator system.

The results of the test are very informative and a tribute to the many people that, during the past three years, have given their best to the design and implementation of the superconducting accelerator. The cryogenic tests were more successful in all respects than what was expected.

This report was edited by G. T. Mulholland and M. I. Martin. We would like to express our appreciation to the Accelerator Division secretaries for their hard work and cooperation in the preparation of this report.

Manuel I. Martin

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I. INTRODUCTION

G. T. Mulholland

The Energy Saver[1] requires 960 superconducting, dipole and quadrupole, magnets organized 40 magnets per Satellite refrigerator, 4 Satellite refrigerators per Sector, in 6, A→F, Sectors total. The powered operation of a full Sector of superconducting magnets to 4000+A was set as a Superconducting Accelerator proof-of-existence test. It quickly became the A-Sector Test, after the Sector location chosen.

A major architect claimed, "The A-Sector Test will operate a sufficiently large segment of the Energy Saver to draw design and operation conclusions valid for the completed Accelerator. It is not a test of a group of components, but the first, *in situ*, test of a significant portion of the accelerator system." The testing and development objectives were as follows:

1. Accelerator component installation and leak checking.
2. Establish and maintain the system insulation vacuum.
3. Cryogenic operation of a Satellite set of refrigerators and compressors, alone, and in combination with the Central Helium Liquefier (CHL) and a mile of helium transfer line.
4. Magnet power supply and quench protection operation spanning multiple subsystems.
5. System local and remote controls, data collection and analysis, and the man-machine interface.
6. Reliability studies of the system under normal operating conditions, and of components and subsystems under specific highly stressed conditions.
7. General operation and experience to establish and train operating crews.

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1. Superconducting Accelerator Design Report, Fermilab, 1979.

The test of six times more 1000 GeV Accelerator magnetic components than had ever been tested before took years of preparation, about four months to complete, and "was more informative and successful than we had reason to expect", according to Peter Limon.

The tests performed come in three subject headings: A-Sector Vacuum, A-Sector Power, and the current effort. The organization of this section is a compendium of individual efforts that provide a lot of the flavor of the test, but little or no uniformity. The reports have not been edited for technical consistency, or form; battlefield notes or polished prose, they are "The A-Sector Test Cryogenic Experience."

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II. STAND ALONE REFRIGERATOR OPERATION

C. H. Rode

During the last several years there have been several stand alone magnet runs: A12 - A17 (25 old dipoles 1979), A21 - A29 (32 old dipoles 1980), B12 (16 old dipoles 1980), A35 - A39 (16 new dipoles May 1981), B12 (16 new dipoles starting May 1981) and A29 - A39 (32 new dipoles Oct. 1981). We had a great deal of problems cooling down, both A3 and B12 during the Oct. 1980 runs with the new magnets; after 3 weeks of effort we succeeded in filling the magnets.

We discovered two problems; the first was expander efficiency roll off with speed; Figure 1 shows that horsepower instead of increasing linearly with speed, leveled off and actually went negative. We made graphs like Figure 1 for both the wet and dry expanders at both B12 and A3. The dry engine graphs were similar to the wet except the roll off was not as dramatic. By slowing the expanders down to the speed half way between maximum efficiency and maximum horsepower (800 RPM for Figure 1) we then easily filled the magnets. It should be noted that in the earlier runs with hydralic loads, we never were able to run too fast since they required horsepower $\geq$ (speed)$^2/c_v$. More current data is given in Section VI by Tom Peterson.

The second problem was high heat load. During the May 81 A3 run we measured the half building heat load at 250 watts. Due to a defective VPT we were not able to separate the valve box heat leak from the magnet heat leak. The Oct 81 A3 run was terminated without taking any heat leak data in order to install the remainder of A-Sector.

I believe the A-Sector heat load was 500 watt per building or 80% of nominal stand alone capacity; see Section VIII by Jay Theilacker for details. This is 180 watts above the old heat leak numbers measured at A1, A2, and B12 on old magnet strings. The Lab 2 spool piece testing measured the new dipole with the 4 anchors and smart bolts at 10 watt (was 7 watts) and the early spool pieces at 18 watts (were ≈ 4 watts); which gives us an increase of 208 watts. (It should be noted that some of the spool piece heat leak has been eliminated for newer units which well make a significant impact on E & F sectors refrigeration loads.

In January 1982, we started stand alone operation. The first A2 cooldown succeeded without operator intervention. On all other attemps, we were only able to fill either string but did not have enough excess capacity to get the second string through transition. After 3 weeks we started up C.H.L. and filled the magnets.

In the later half of the run we turned off the liquid He from C.H.L. on three separate occasions. The first time CHL went down and we shut off liquid to A2 and A3 running A1 off of the transfer line. The length of this run was 6 hours which was too short to reach complete equilibrium. The second run attempt we reached equilibrium at A2 and A3 but not at A1.

The third was a 35 hr run in which we cut the magnet JT flow down to that required by the heat load and measured the total coldbox helium flow. We achieved about 1.0 compressor equivalent flow at A2 and 0.9 at A3. This corresponds to 560 watt load with and without power leads. At A1 we ran in stand alone for 9 hours, but again did not reach equalibrium.

To Summarize:

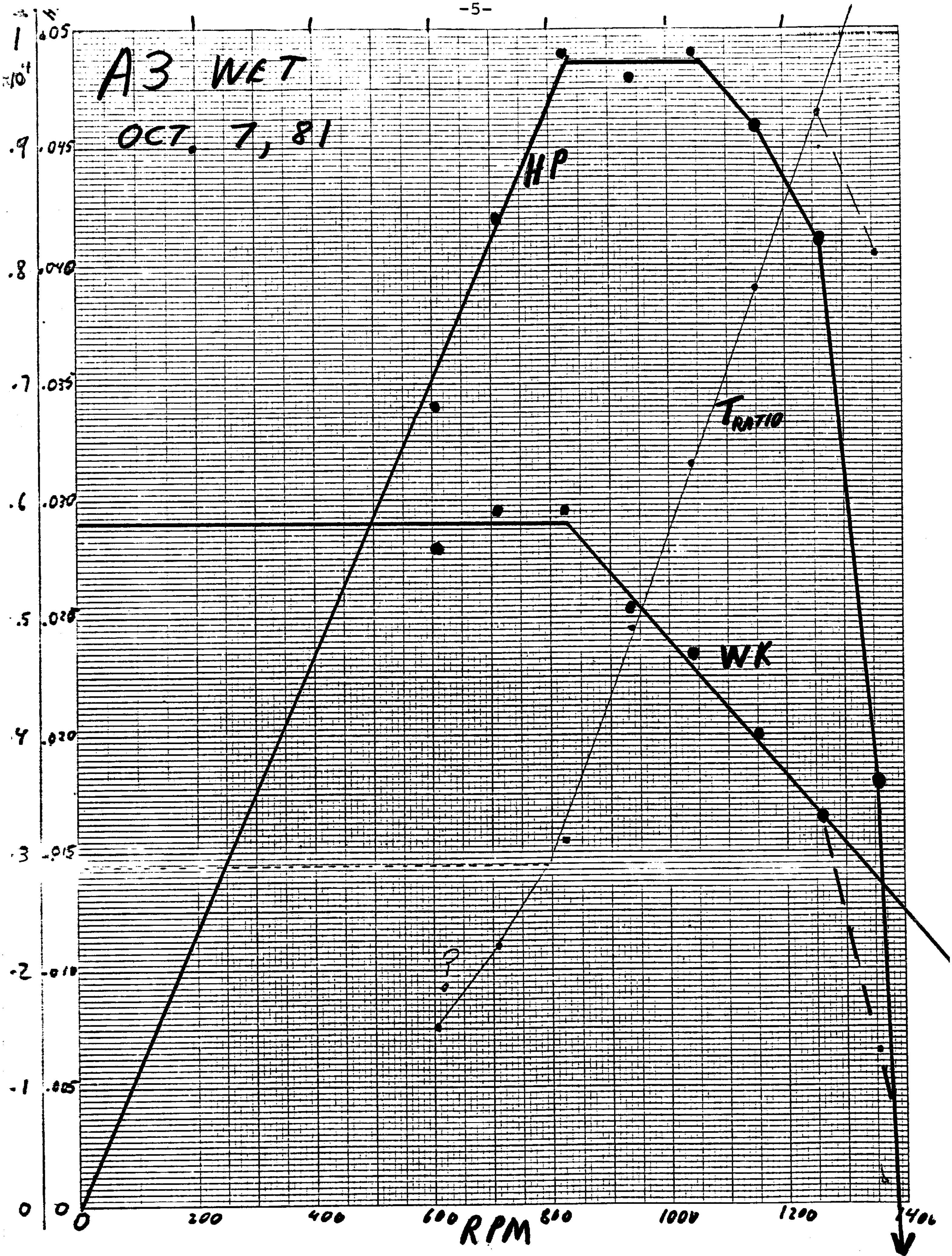
1. Until the expander efficiency roll off problem is solved, we cannot fill the magnet reliably in stand alone mode. We can easily cool them down to 25K; then we need CHL to fill them.

2. Once the magnets are full we can easily keep them full of liquid. The main weakness is that it requires 100% of our compressor capacity; we are therefore adding 3 additional compressors at B0 which will give us 13% extra.

FUTURE EXPERIMENTS

1. We must understand the heat load of the magnet; during cooldown we will do the simple single pass heat leak measurements. After filling the magnet we will do loop heat leak measurements using both the warm and Cryogenic flow meters. See Section VIII by Jay Theilacker. We must know what the heat loads are in order to measure the efficiency of the refrigerator. If the heat load is 500 watt all we can do is work on expander roll off. If the heat load is 375 watt we have a serious loss of efficiency that must be found.

2. We must understand what is causing expander roll off. All experiments to date have only shown that we have a problem; we need to design experiments that pin point the cause.


A3 WET
OCT. 7, 81
HP
WK
T RATIO
?
RPM
0
200
400
600
800
1000
1200
1400
.005
.010
.015
.020
.025
.030
.035
.040
.045
.05
.1
.2
.3
.4
.5
.6
.7
.8
.9
1
$\times 10^{-4}$

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III. CENTRAL HELIUM LIQUEFIER OPERATIONS

R.J.Walker

Preparations for the A-Sector test started in June, 1981 with the installation and testing of the Central Helium Liquefier subcooler and the transfer lines to A-3 and A-4. Testing of the subcooler was completed with LN_2 flow at expected operating conditions by dumping the correct flow of about 2000ℓ/hr overboard. Tests of flow control and heat leak measurements proceeded until Sept. 3, 1981. The heat leaks were close to design values and the control was satisfactory. In order to establish satisfactory flow, control, and heat leak, it was necessary to flow cold helium from A-3 to the CHL, through a jumper, and on to A-4. A helium flow of the order of 25 liquid liter/hr was used. LN_2 flow was of the order of 3000 scfh and was warmed and vented outside at A-3. The flow of helium was necessary to maintain good insulating vacuum. If poor vacuum developed, the nitrogen pass of the transfer line would develop gas pockets which would cause the line to surge forward and alternately flow back, and subcooling could not be maintained.

A cooldown of the CHL began on Oct. 5, 1981 to check the readiness of the system to perform the A-Sector test. At this time the status of the system was as follows:

1. The cold box and compressor system had been tested.
2. Gas cleanup and monitoring systems were satisfactory.
3. The transfer lines to A-3 were installed and leak checked.
4. The nitrogen subcooler was installed and tested.
5. The helium distribution box and subcooler were installed but not tested.
6. A new system of controls for the heater had been installed and tested.
7. Protective devices were installed on the 5kV switchgear to prevent failure in the disconnect switches.

The run proceeded well with the cold box and compressors. The testing of the helium subcooler indicated that the piping had an internal error. The run was terminated on Oct. 15, 1981, to correct the problem in the subcooler.

The subcooler was cut apart and the internal piping was fixed. A cooldown began on Oct. 23, 1981. The cooldown was normal. The subcooler was tested with helium gas into the subcooler. The 5kV switchgear malfunctioned frequently when starting compressors. The Texas Instruments TI5000 which controls the compressor quit without tripping the compressor. The run was terminated on Oct. 25 to fix the problems with the switchgear. All new components of a more rugged type were installed. No problems with the switchgear have been experienced to date (July 26, 1982).

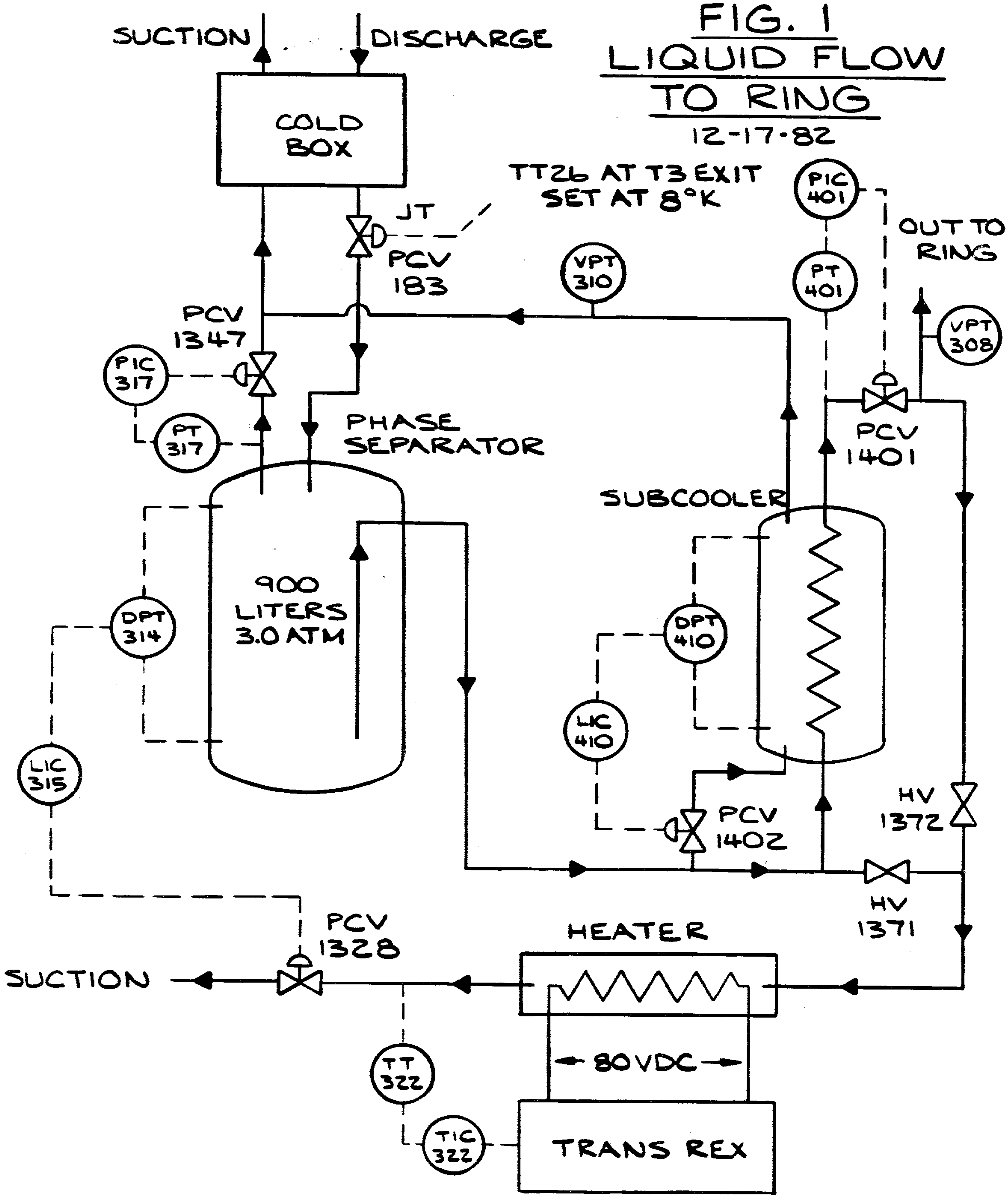
The third cooldown of the series began on Dec. 7, 1981. The main objective of the run was to gain experience in sending cold helium at supercritical pressure to the Ring. Figure 1 shows the main flow system to route the liquid helium to the Ring.

The system works as follows. Fluid flows through the JT valve into the phase separator. A normal JT valve drops the pressure to a subcritical pressure and two phase helium is formed. In the present situation, the pressure drops only to 3 atm and no liquid is formed. For the liquefier to work correctly, the proper amount of cold fluid must return through the heat exchanger. The correct split of about 50% is performed automatically by PCV-1402 which takes just enough flow to keep liquid in the shell side of the subcooler. The boil-off from the subcooler returns through the low pressure side of the cold box and gives the correct balance of flow.

The remaining fluid flows through the tube side of the subcooler and either to the Ring or to the heater. PCV-1328 is a room temperature pressure control valve which controls the flow into the heater. It regulates the differential pressure between the top and bottom of the phase separator. This control actually controls the temperature of the phase separator. This temperature must be maintained at a sufficiently low temperature or no liquid can be formed in the shell side of the subcooler.

Since PCV-1328 maintains the phase separator at the right temperature it automatically compensates for changes in flow to the Ring and permits the excess capacity to go into the heater. PCV-1347 main-

FIG. 1
LIQUID FLOW TO RING
12-17-82



tains the phase separator and the transfer line at the correct pressure. Only a small flow is taken by PCV-1347, and the main split is determined by PCV-1402.

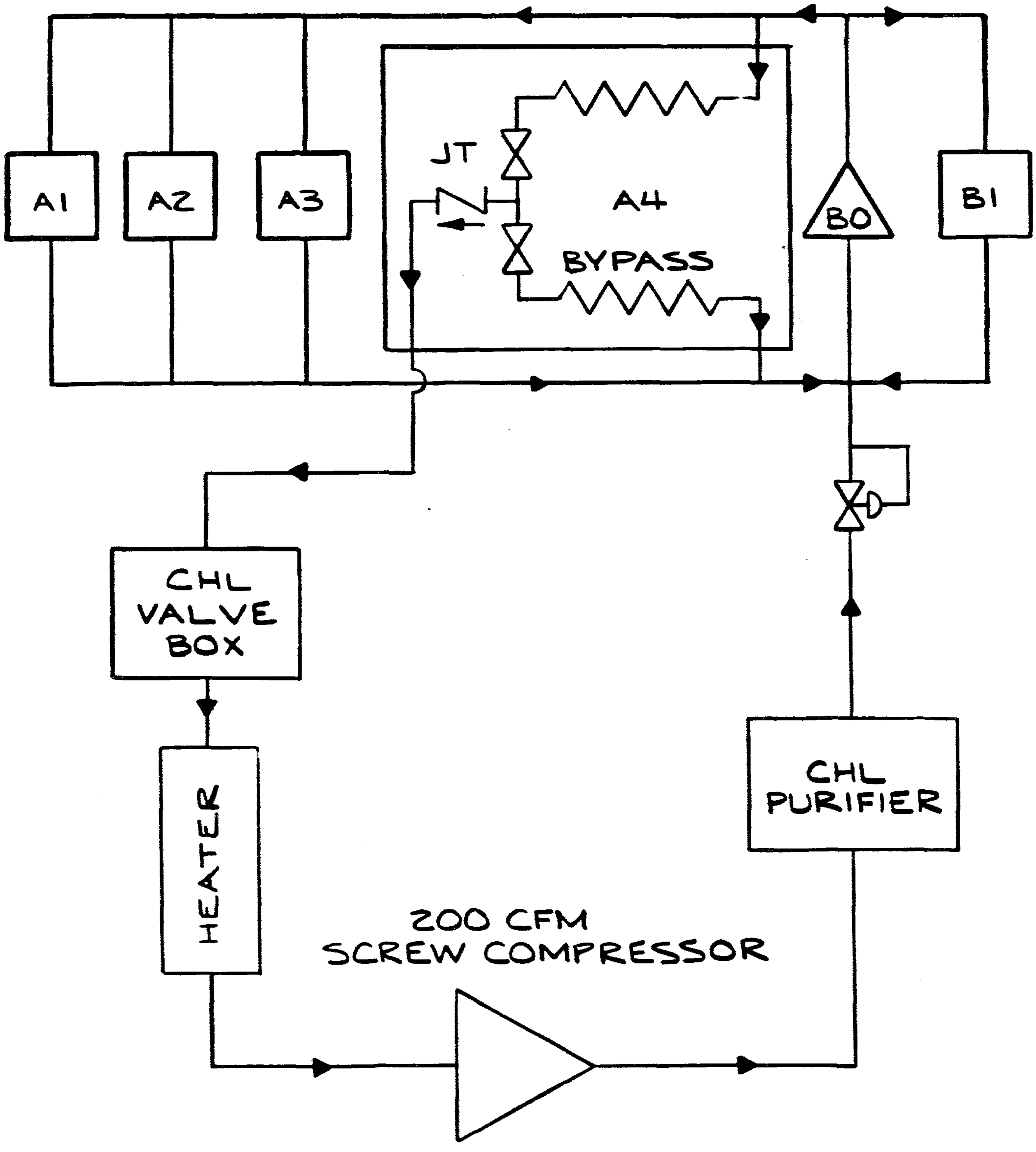
During this run the CHL ran very smoothly, and the crew moved rapidly up the learning curve. A problem developed with resonant shaking of the dipole magnets nearest the CHL, and the CHL was required to shut down one compressor until the Ring magnets were sufficiently braced. The CHL was performing at least at design production, and a peak capacity of 3800ℓ/hr was observed with no return flow from the Ring. Preliminary testing of the transfer line was accomplished by sending 45ℓ/hr to A-4. Supercritical operation of the phase separator was successful.

After the CHL warmup, the purifying system was used to provide scrubbing service to A-Sector. Figure 2 shows the flow path used. This down-time was also used to pipe in two new 30,000 gal buffer tanks. The tanks could not be adequately decontaminated during the cold weather, but they could be used for dirty gas storage. Gas was passed through a drier and the CHL purifier before returning it to the system.

The fourth cooldown of the series began on Feb. 1, 1982. The main progress during this run was the successful operation of the transfer line and the filling and maintaining of liquid helium in the magnet strings so that power tests could proceed. The main difficulty with this operation at the CHL was the surging behavior of the transfer line, which required almost constant operator attention to prevent severe plant upsets. The plant can cope with the outward surges of the line, and the heater controls simply give the full capacity to the Ring. If the surge is too big, however, the phase separator empties and the plant begins to warm up and send warmer fluid to the Ring. In general this effect did not upset the Ring. If the flow reverses, the situation is much worse for the CHL, and the reverse flow causes the CHL to go bananas. We have not to date found a way to cope with the line sending liquid back to the CHL, and we believe the only good solution is to have a liquid dewar to take this back-flow and enable the CHL to run in steady state.

During the February run, the CHL was plagued with problems with cooling tower gear boxes and failing valves in the second stage of the main compressors. The main diffusion pump on the cold box also had a cooling problem. We were able to fix these problems without shutting

FIG. 2
SCRUBBING FLOW TO RING
1-5-82



down. It was obvious that a major modification was needed in the second stage valves, and this modification was made after the run. The Texas Instruments TI-5000 also required numerous replacements of failed components. The run was terminated on Mar. 23, 1982 after a single phase rupture at A-15 caused by an intentional quench.

During the shutdown, the major effort was to rebuild all the second stage valves in both compressors. The ends of the valve strips were padded with Teflon buttons. This repair has improved the compressor performance greatly and no compressor problems have developed in the last 1200 hr of operation. The new buffer tanks were properly decontaminated, and a number of system improvements were made.

The fifth cooldown of the series began on Apr. 18, 1982. The system behaved very well during this run. The main problem was transfer line surges which required vigilance to prevent excursions of the plant to some trip point, either in temperature or low compressor discharge. Efforts were being made to stabilize the long transfer line, which ran all the way to E-0. These were fairly successful when pressure regulation was attained by regulating the flow from the helium transfer line to warm suction at E-0. This fix seems to prevent the transfer line from surging as long as the demand by the satellite refrigerators did not change. There were still sufficiently large back surges to upset the CHL whenever the demand change caused the line pressure to go above the supply pressure at the CHL.

During the preceding runs, the noise level in turbine 1 seemed to be too high. The Sulzer representatives were asked to come and help with the problem. After the Sulzer people had a chance to listen to the turbines in operation, the plant was warmed to room temperature and the turbines were removed and disassembled for inspection. There was considerable dust from the heat exchangers in the impeller area, and T1 inlet filter was damaged. This sort of dust is a general property of the Trane heat exchangers and did not damage the turbines. The oil bearings and brakes of T2 and T3 were judged to be ok. T1 brake rotor and the matching part of the cartridge were eroded by the oil flow, and the rotor and housing are being replaced along with the bearings.

The system was reassembled with a new T1 inlet filter and spare parts in T1 cartridge. During a test run with dummy plates in place of the turbines, a high noise level was observed in the turbine inlet and outlet pipes. These pipes were wrapped with 3/16" thick lead strips to damp the vibration. The oil in the turbine oil skid was replaced and new oil filters were installed. The turbine system has been purged to helium gas and is ready to run for the next testing period.

In conclusion, it is desirable to point out that operationally the CHL worked very well, and the crew did a good job in maintaining high efficiency during the two long runs while power testing was proceeding. The run from Feb. 3 to Mar. 23 lasted for 1143 hr. Liquid was delivered to the Ring for all of this time but 15 hr. This represents an efficiency of 99%. The run from Apr. 19 to June 11 lasted for 1216 hr with liquid flow off for only 14.5 hr. This also represents an efficiency of 99%. The longest interruption in liquid flow was for 7 hr, and the inventory in the transfer line was able to keep the satellites and magnet strings from warming up until the CHL came back up. The satellites went into a stand-alone mode during these intervals.

<u>Chronology of Events at the CHL During the A-Sector Test and Preparations</u>

6-15-81	Test liquid nitrogen subcooler.
7-10	LN_2 flow to A-Sector.
9-3	Transfer line heat leak measurement to LN_2.
10-5	Start CHL cooldown.
10-13	Valve box defective.
10-15	Start CHL warmup.
10-22	Valve box fixed.
10-23	Start CHL cooldown.
10-25	Warmup CHL.
12-2	Installing new switchgear controls, replacing Slypsyn with Baesler.
12-7-81	Start CHL cooldown.
12-9	Ring magnets shaking, go to A compressor only.
12-13	Turbines noisy.
12-15	Stable supercritical flow to heater.
12-16	PPM oil monitor readings: 27mg/m^3 at discharge; 124μg/m^3 after demister; 0 after adsorber
12-17	Palmer says Ring not shaking with two compressors running, Ring magnets adequately braced.
	Flowing 45ℓ/hr subcooled liquid helium to A-4 Service Bldg.

12-17-81	Try running T3 inlet at 7.5K instead of 8.0K, turbines noisy, return to 8.0K.
	Plant capacity 3800 /hr, two compressors + LN_2, no Ring return flow.
12-18	Oil monitor cross check between Balston and ppm.
12-19	Warmup CHL.
1-5-82	Scrubbing flow to A-Sector.
1-7	Install two 30,000 gal buffer tanks, temporary hookup.
1-30	Purge and scrub new buffers.
2-1	Start CHL cooldown.
	Gear box on B cooling tower fan failure.
2-2	Gear box replaced.
	Area oxygen monitors installed.
2-3	Bad valves in B second stage.
	Phase separator burst disc failure, defective burst disc. Replace burst disc.
2-4	Filling A-Sector.
	Liquid helium at A-2 in 2 hr.
	Area oxygen monitors operational.
2-5	B compressor vibration trip.
	Replacing valves in B second stage.
	B compressor up, scrubbing.
	B compressor on line.
	Leaking stinger on valve box due to thermal oscillation. Reduce phase separator pressure to 2.8 atm to reduce oscillation.
	Transfer line oscillations, putting gas in new buffers.
	Scrubbing gas from new buffers.
	CHL stable except for transfer line burps.
	Lower phase separator pressure to empty transfer line. Satellites can't get enough helium out, pressure up.
2-11	B compressor down, second stage vibration. Repairing B second stage. Piston has lost crush. Machining at Commercial Machine Works.
2-15	B compressor up, scrubbing.
2-16	T1 inlet filter plugging coincident with B-12 warmup. 10.6 psi differential.
2-18	CHL trip on instrument air interlock.
	Warm T1 filter to -66C.
	Liquid to Ring in 5 hr. T1 filter clear.
2-24	Main diffusion pump trips on hi-temp, purging water circuit with citric acid.

3-1-82 Ring trying to identify source of steady 7 ppm N_2 in return gas. No luck.

3-7 B compressor down, second stage valves.

Valves replaced, B up, scrubbing.

3-11 B compressor down, try different velocities in second stage valves.

3-13 B compressor on line.

Switchgear trip, off 2 hr.

T1 noisy, took accelerometer readings.

3-17 Shut down CHL to clear T1 inlet. Ring can hold. Take sample during warmup.

Liquid helium in 5 hr. Ring ok.

3-18 W.Robertson of Koch in to listen to turbines.

3-21 CHL trip, vibration switches lost power. Back up in 5 hr.

Another velocity change in B second stage valves.

3-22 Single phase rupture at A-15.

3-23 Warmup CHL.

3-26 Drying new buffers.

3-31 Observe sequencer latch up.

4-16 Rebuild all second stage valves with Teflon buttons.

Improve systems: Transrex OFF alarm
Heater cryoline bellows
More switchgear readouts
New LN_2 level gauge
Pulsation dampers on oscillating cryolines
More new parts in sequencer

4-19 CHL cooldown.

4-22 Transfer line burp trips CHL.

Pressure oscillations in transfer line continue.

4-29 Back pressure regulator at E-0 should stabilize transfer line flow.

4-30 New buffer valves wired to operate from the Control Room.

5-11 Continuing plant upsets correlated with helium coming back from Ring.

All agree to maintain fixed restriction at CHL end of transfer line.

5-12 CHL upset and trip. Probably related to transfer line. Back up in 2.5 hr.

CHL down. Sequencer died. Replace input-output expander and one interface module. Up in 4 hr.

5-18 CHL trip, vibration switch power. Slow cleanup, back in 7 hr.

5-20 Replace B compressor cylinder oiler.

5-23-82	Noise in gear box in B cooling tower. Found oil plug with drilled hole. Will change gear box.
5-24	Plant abnormally stable.
6-2	Routine blowdown of purifier inlet filter. Strong odor.
6-6	Loss rate measurements when systems are stable.
6-10	Plant and transfer line stability continues.
	Close liquid supply to Ring. Ring emptying.
	Total gas back from Ring, 100912 scf.
6-11	Warming CHL to derime T1 inlet filter.
	Tracor says 80 ppm Vol. neon in process during warmup.
6-12	Cooling down CHL.
	CHL only loss measurements.
6-15	Sulzer representatives arrive to help with turbine inspection.
	Warmup CHL.
6-16	Turbines out, dummies in, cleaning dust from process.
6-25	Repairs and cleanup complete. T1 spare cartridge with repair rotor set installed. T2, T3 installed without change. T1 parts sent to Sulzer. New T1 inlet filter installed. System leak checked with helium sniffer. Turbine inlet and outlet pipes wrapped with 3/16" lead. New oil and oil filters in oil skid. System purged to helium. Sulzer representative returns to Fatherland. End of A-Sector test.

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IV. TRANSFER LINE OPERATION

J. Makara

At the start of A-sector test, cooldown of the magnet strings was attempted in stand-alone mode, with only the satellite refrigerators providing the required refrigeration. After several days of attempts to fill six magnet strings full of liquid helium, the Central Helium Liquefier (CHL) came on line and by means of the transfer line (TRL) provided the additional boost to complete the filling. This so called satellite mode of operation provided an opportunity to observe TRL interactions during various conditions of operation.

Initial TRL cooldown involved A3, A2, A1 to F4 only. At F4 a connection was made to a return warm line (natural heater) back to AØ suction, Figure 1. Flow rate was controlled at each operating refrigerator (A1, A2, A3) as required by that refrigerator through electric valves (EVLH) and at AØ by manually controlled flowmeters. CHL provided fast cooldowns and fillings of magnets after quenches or such. Extreme pressure surges in the TRL were experienced during most of this run sequence. These (approximately 6 hour) oscillations caused not only CHL operation problems but pressures up to 50 psig (nominal operation is 24 psig) opened two TRL relief valves several times, loosing some helium inventory and requiring extensive maintenance in defrosting and reseating the valve. There were eventually replaced for a more recent improved design. This TRL section ran for two months with CHL anticipating pressure spikes and closing their supply valve to dampen the effect on their operation. Many power tests were performed on the magnet strings neverthless.

Sectors F and E transfer lines were added to the cryogenic system in the hopes of damping the TRL pressure oscillations and also to obtain a measurement of the heat load on the helium part of the TRL. The added line also provided a source of liquid helium for quick quench recovery or temporary supply during CHL shutdown, in a sense a 3200 liter helium dewar. Figure 1 showed the E1 connection to AØ suction. Oscillations persisted with extreme negative spikes every 18 hours and positive ones every 10 hours. Figures 2 and 3 show the pressure oscillations and temperature profiles in E and F sectors, respectively.

One reasonable explanation for the oscillation is as follows. First, the TRL is filling very fast, verified by both the CHL usage and the low TRL pressure. The low TRL pressure is caused by the condensing warm helium gas ahead of the cold dense helium supply from CHL. The input mass flow is significantly greater at the inlet as compared to the outlet warm gas at E1. When the cold wave finally reaches E1, flow restrictions in the U-tube flowmeter, the flashing of dense gas in the return line to AØ plus a major restriction at AØ flowmeters produce a significant back pressure to decrease the flow through the TRL to a point that some backflow occurs at the inlet and a region of the TRL becomes stagnant. This stagnant region warms up, growing a bubble of warm gas until the fluid (actually dense supercritical gas) downstream toward E1 is emptied at E1. The fill cycle again begins, repeating the cycle.

Additional outlet flow at F4 was attempted though its affect is inconclusive. Temperatures at F3 and F4 did stabilize in time. CHL attempts to control the fill cycle by throttling back on their supply valve to keep TRL pressure up was unsuccessful. Shortly afterwards, an electric valve control was added to E1 to provide TRL pressure regulation. That control, together with a flow-limiting orifice at CHL, significantly stabilized pressure oscillation, Figure 4 shows that improved situation. Instrumentation was also added to E1 to read TRL temperature. Temperature oscillations to exist at E1, Figure 5 is a typical depiction.

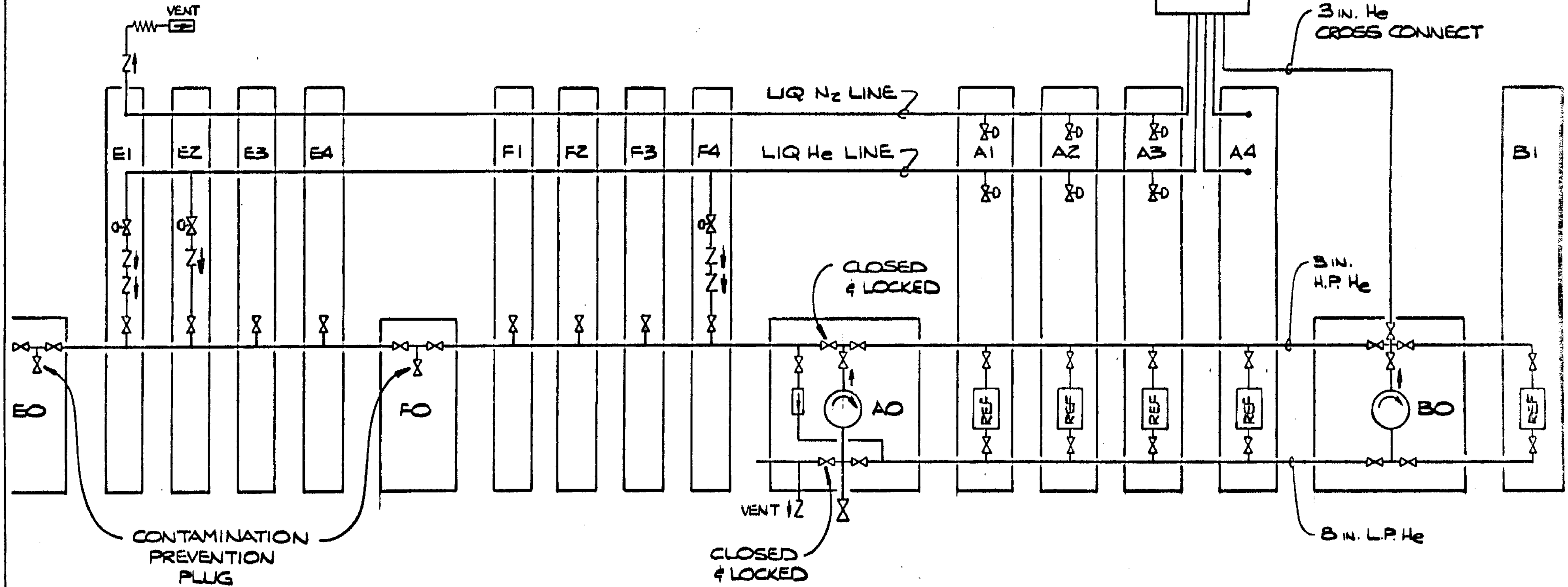
To take advantage of CHL's full output, AØ flowmeter capacity was increased. Modifications were also made to allow outlet flow at E2. Shortly afterwards, TRL termperature oscillations decreased, Figures 6 and 7, though not enough to allow an accurate measurement of the heat load. The interaction of refrigerator operations and TRL behavior can be seen in Figures 7 and 8, not only in quench recovery, but oscillations of EVLH at the refrigerator.

In conclusion, TRL oscillations that existed did not hamper general operation of the three refrigerators at A1, A2, and A3. As more refrigerators come on line, oscillations should diminish due to the increased outlet flow capacity. Near the end of the A sector tests, TRL oscillations had greatly diminished (Figure 9) and remaining disturbances can be attributed to A sector operation conditions such as quenches or valve oscillations.

REV.	DESCRIPTION	DRAWN / APPD.	DATE / DATE

FIGURE 1

HALF RING TRANSFER LINE TEST - APRIL '82



UNLESS OTHERWISE SPECIFIED	ORIGINATOR	C. RODE
FRACTIONS ± DECIMALS ± ANGLES ±	DRAWN	D. RICHARDSON 3-30-82
1. BREAK ALL SHARP EDGES 1/64 MAX. 2. DO NOT SCALE DWG. 3. DIMENSIONING IN ACCORD WITH ANSI Y14.5 STD's. MAX. ALL MACHINED SURFACES	CHECKED	
	APPROVED	
	USED ON	
	MATERIAL-	

FERMI NATIONAL ACCELERATOR LABORATORY
U.S. DEPARTMENT OF ENERGY

SAVER CRYOGENIC SYSTEMS GROUP
HALF RING TRANSFER LINE
TEST SCHEMATIC

SCALE	FILMED	DRAWING NUMBER	REV.
~		1650-MC-167092	

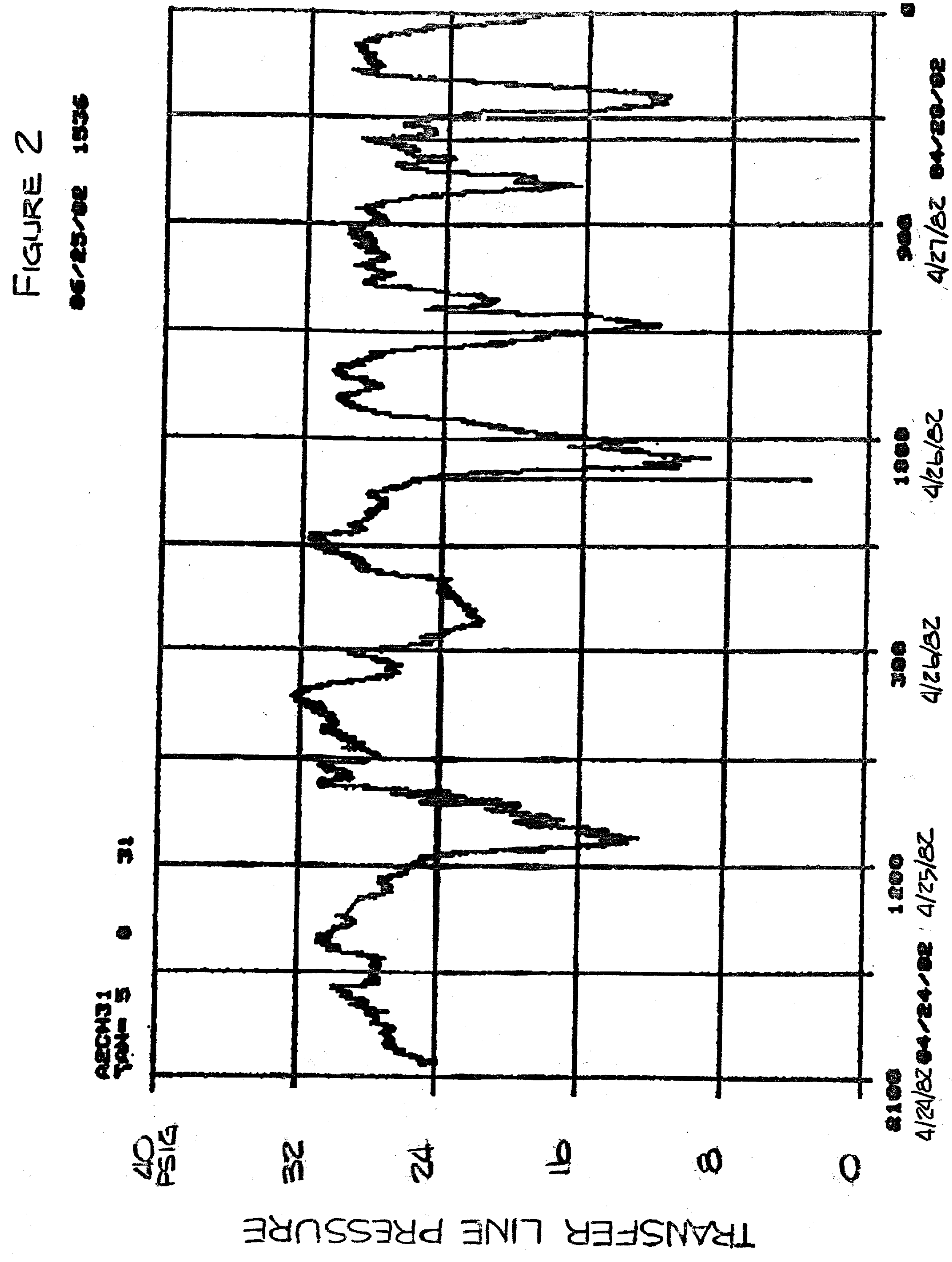


FIGURE 2
06/25/82 1536
AECM31
TAN= 5
0
31
TRANSFER LINE PRESSURE
40 PSIG
32
24
16
8
0
2100 4/24/82
04/24/82
1200 4/25/82
300 4/26/82
1800 4/26/82
900 4/27/82
04/28/82

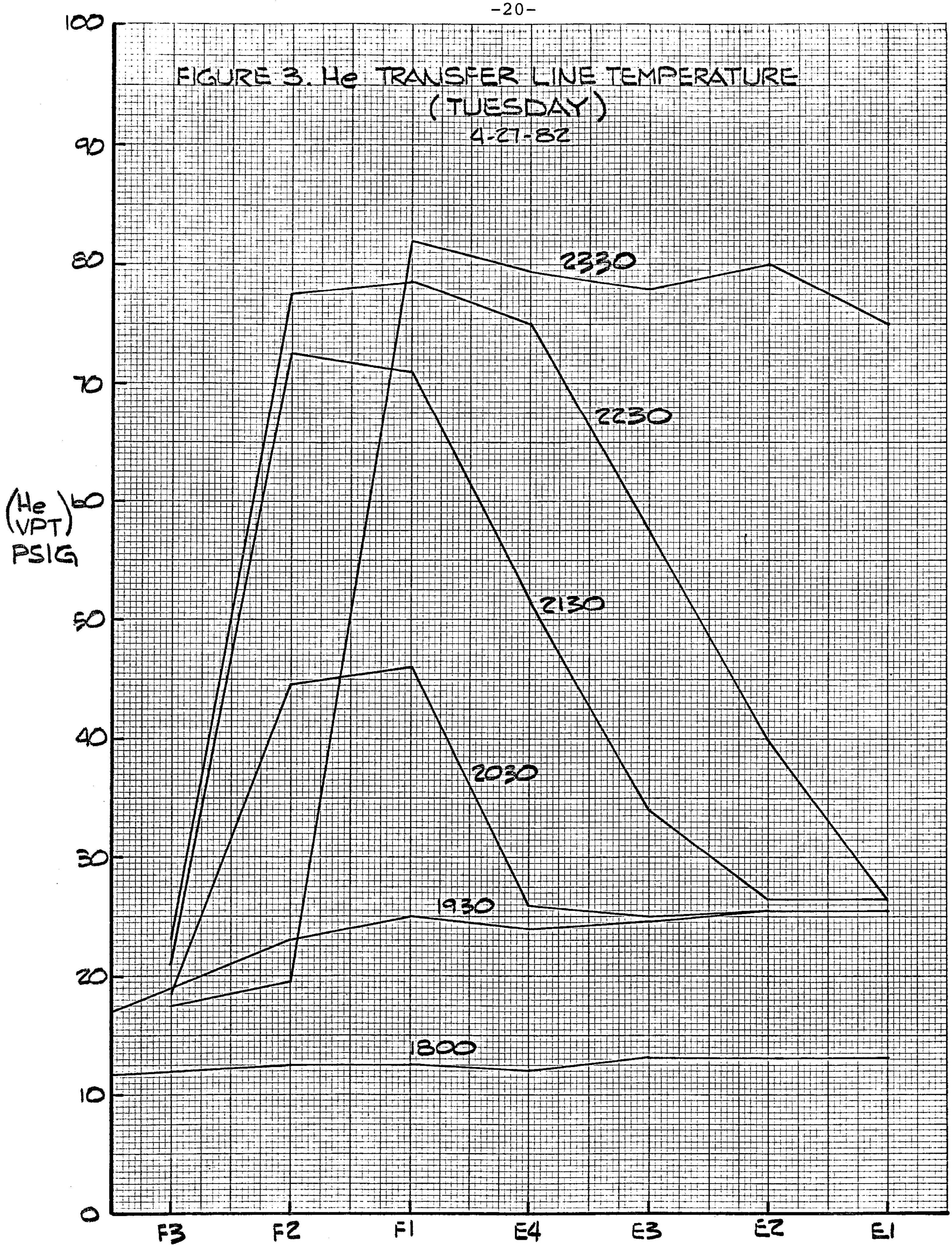


FIGURE 3. He TRANSFER LINE TEMPERATURE
(TUESDAY)
4-27-82
100
90
80
70
60
50
40
30
20
10
0
(He VPT) PSIG
2330
2230
2130
2030
1930
1800
F3
F2
F1
E4
E3
E2
E1

SCALE: TRL PRESSURE 0-40 PSIG
E1 HELIUM TEMP. (CARBON RESISTOR) 0-100Ω

FIGURE 4

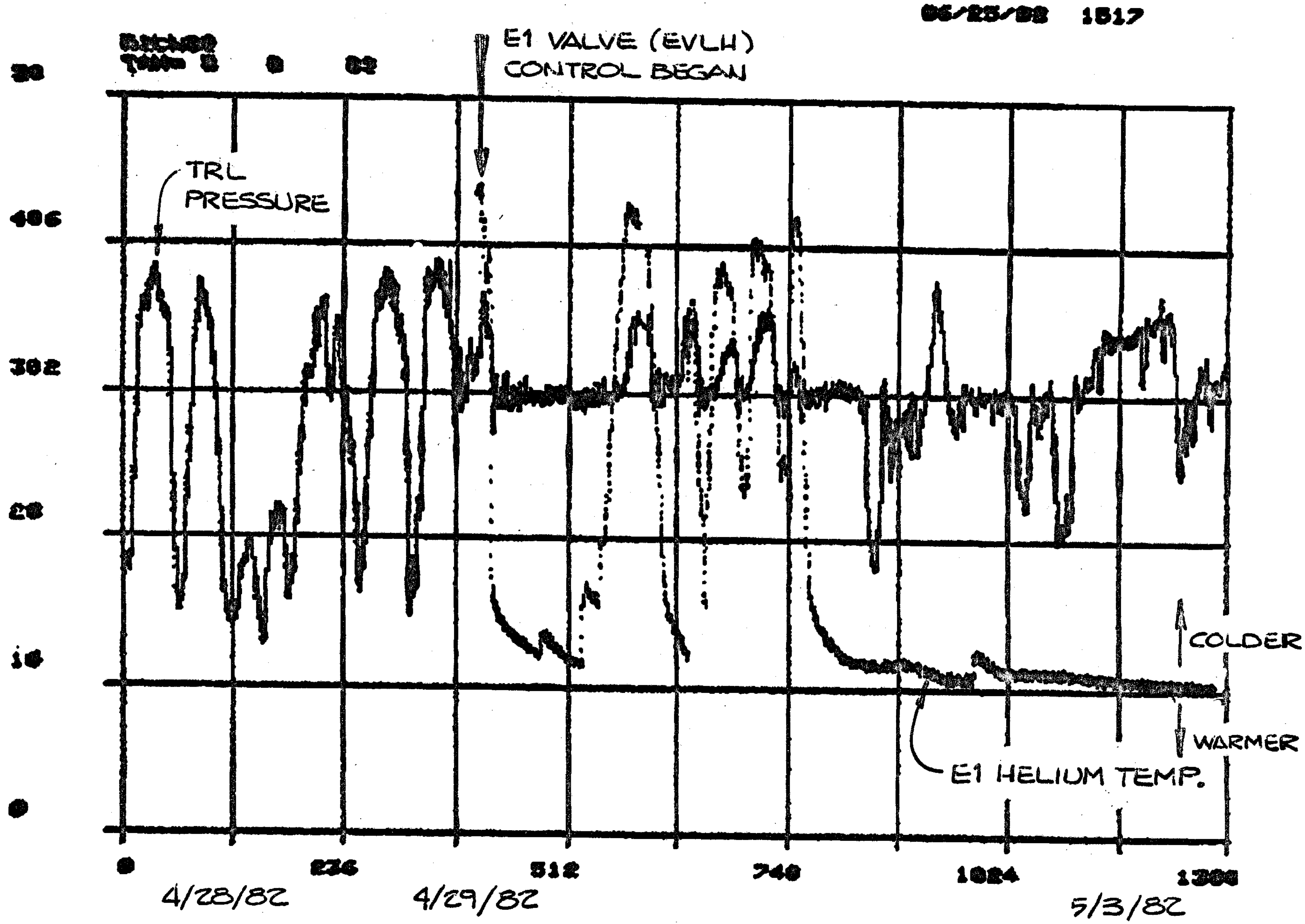
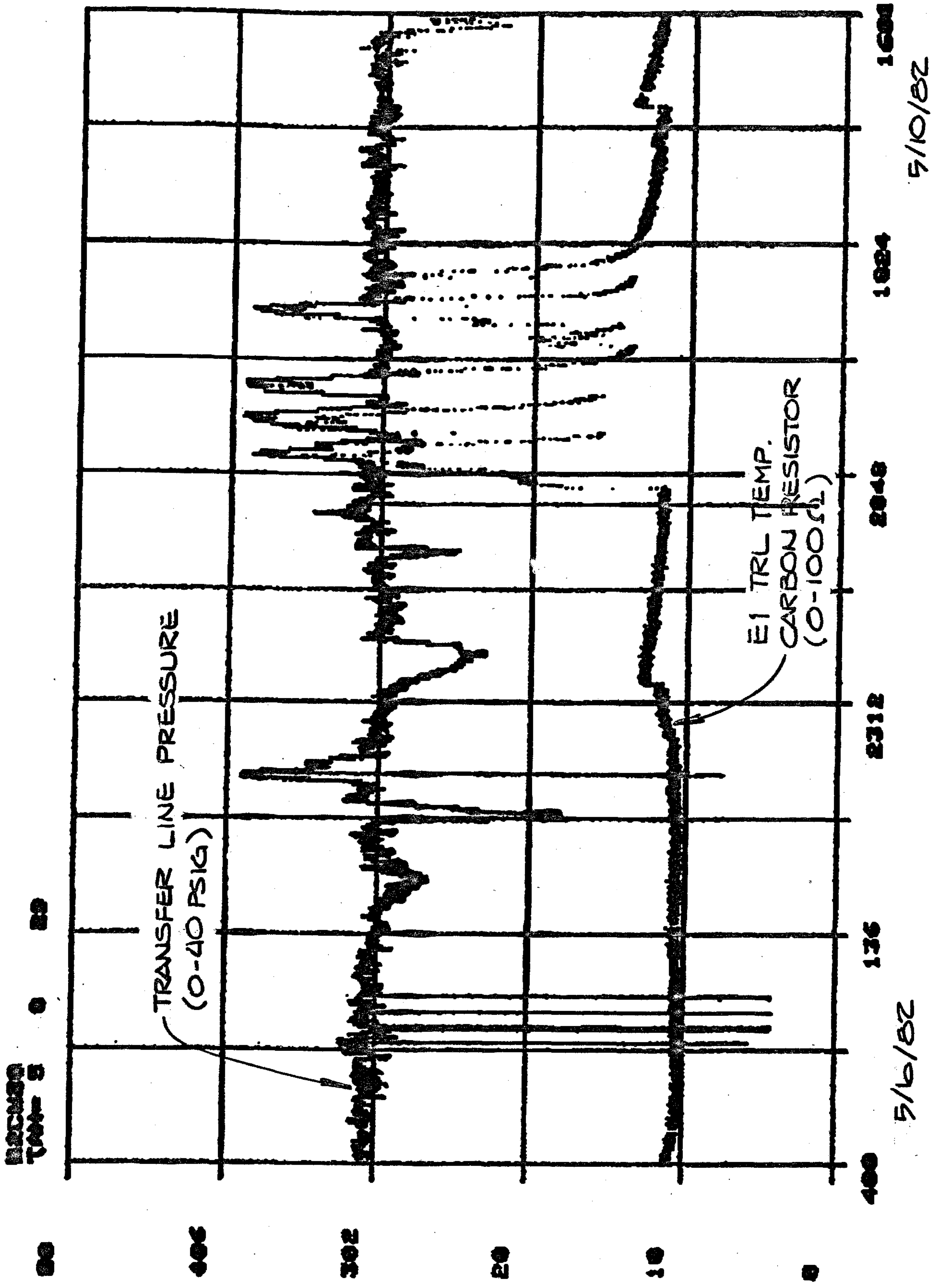


FIGURE 5


TRANSFER LINE PRESSURE
(0-40 PSIG)
E1 TRL TEMP.
CARBON RESISTOR
(0-100 Ω)
5/6/82
5/10/82
136
2312
2048
1024

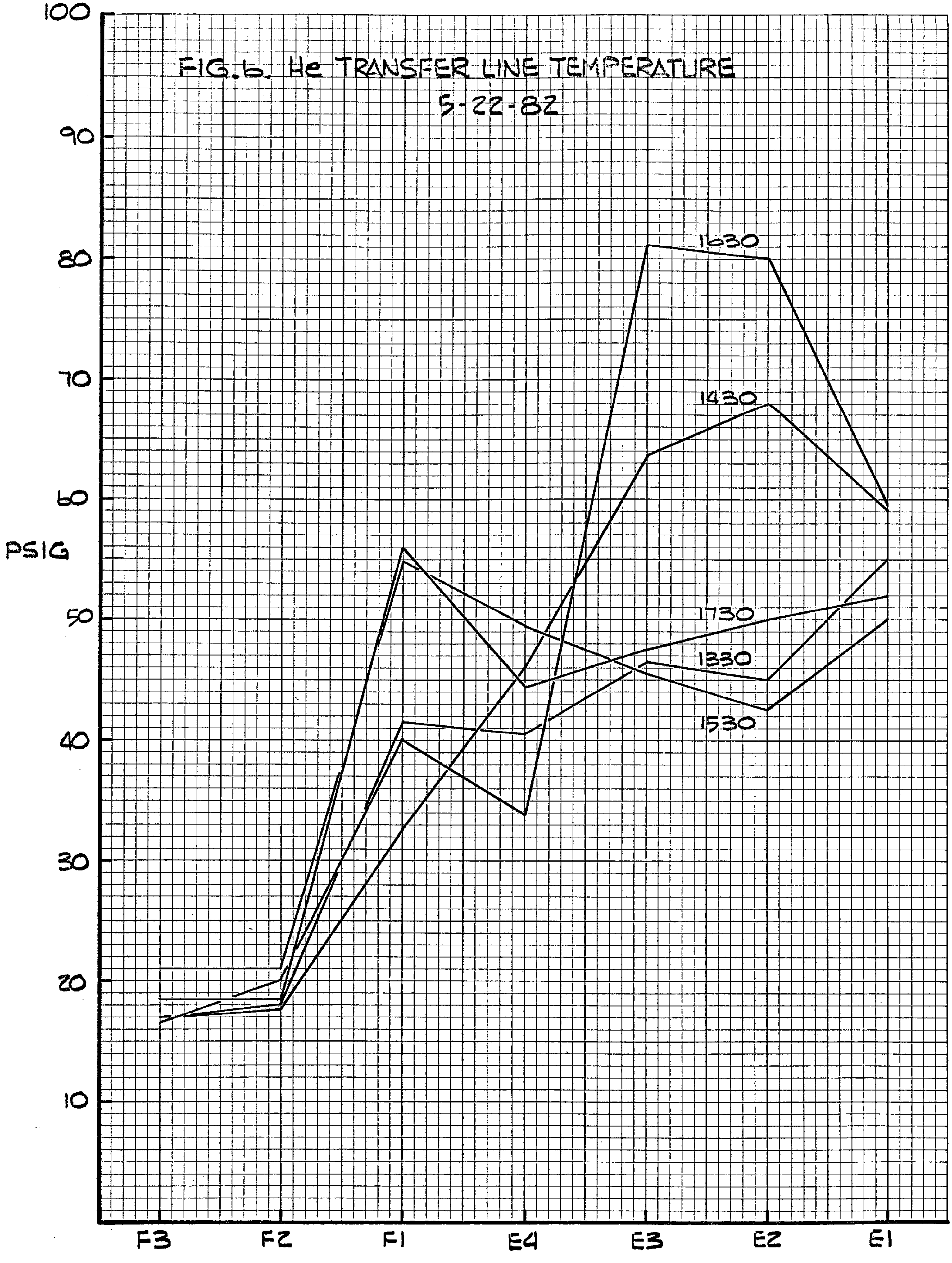


FIG. 6. He TRANSFER LINE TEMPERATURE
5-22-82
PSIG
100
90
80
70
60
50
40
30
20
10
1630
1430
1730
1330
1530
F3
F2
F1
E4
E3
E2
E1

FIGURE 7

06/24/82 1948

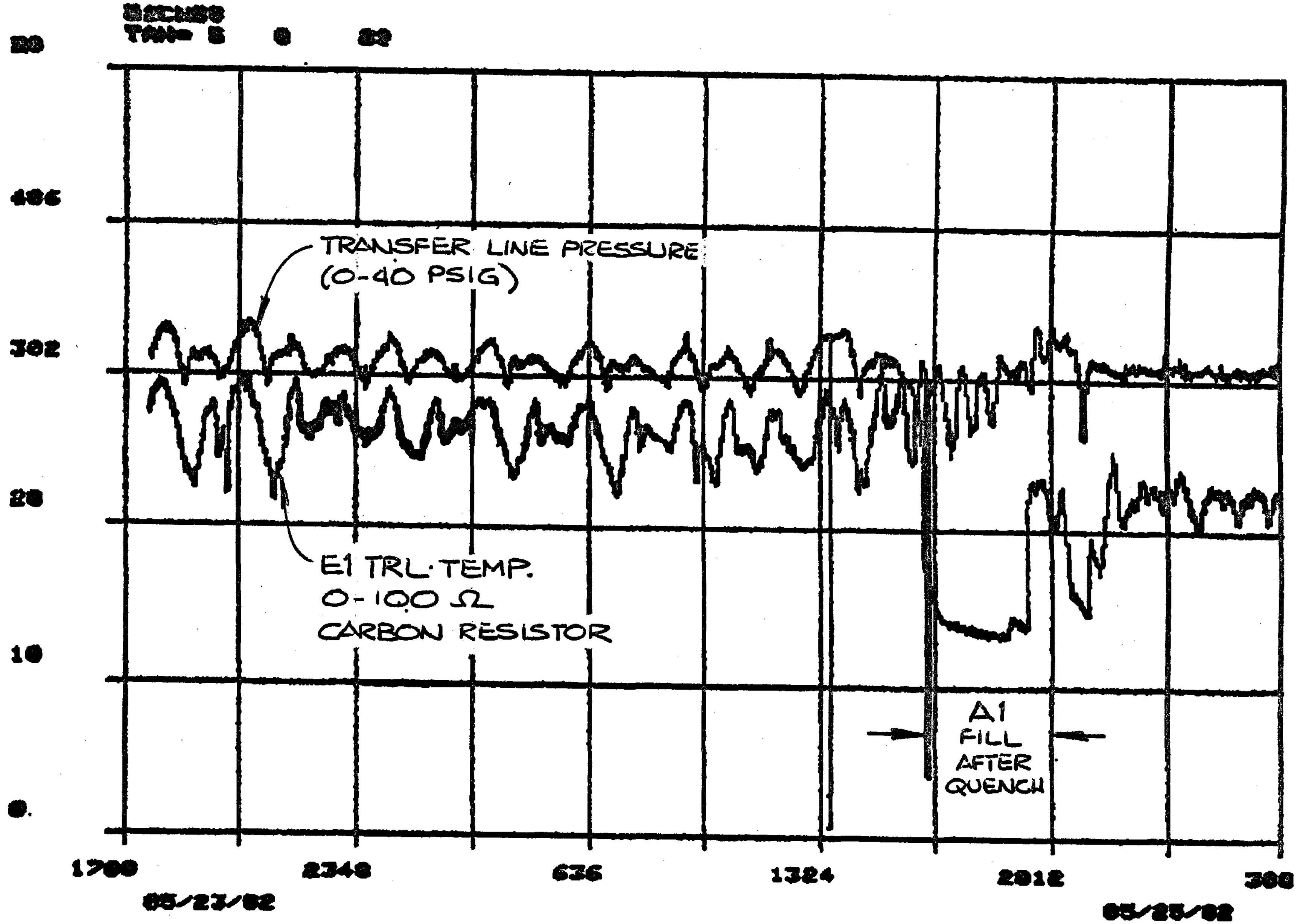
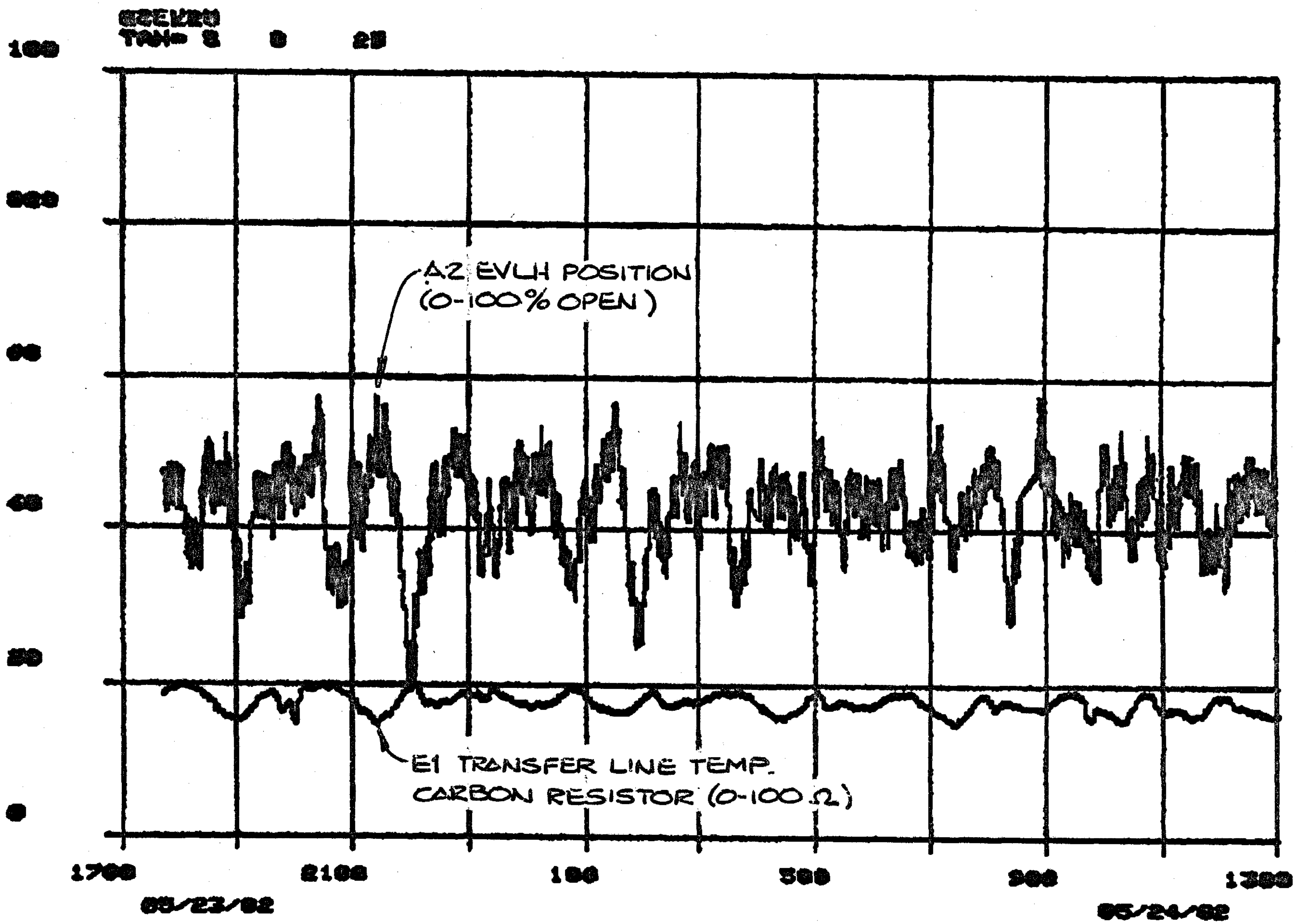
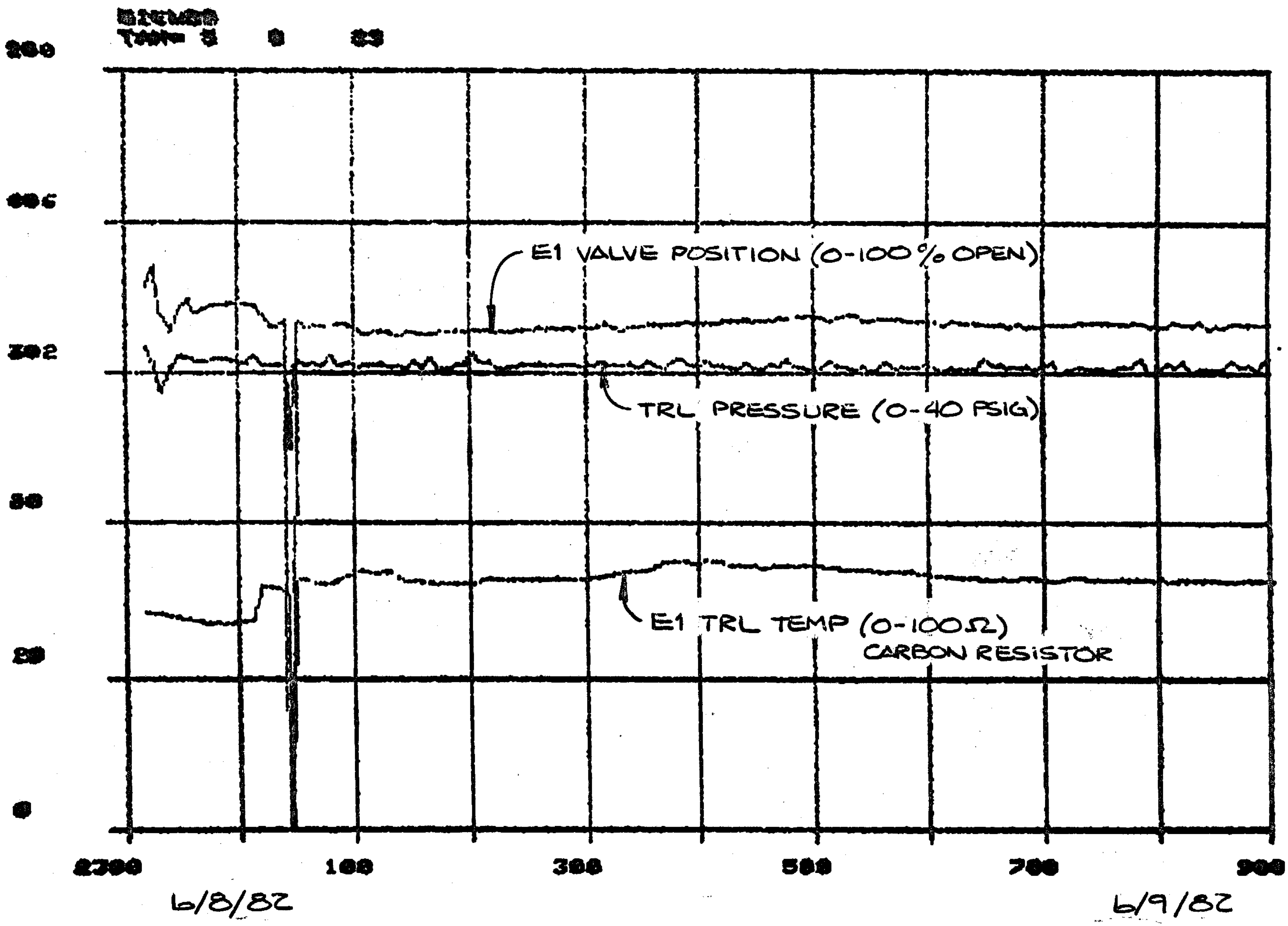


FIGURE 8


A2 EVLH POSITION
(0-100% OPEN)
E1 TRANSFER LINE TEMP.
CARBON RESISTOR (0-100 Ω)
1700
2100
100
500
900
1300

-25-

FIGURE 9


E1 VALVE POSITION (0-100% OPEN)
TRL PRESSURE (0-40 PSIG)
E1 TRL TEMP (0-100Ω)
CARBON RESISTOR
2300
100
300
500
700
900
6/8/82
6/9/82

V. COMPRESSOR OPERATION

M. Hentges/J. C. Theilacker

Introduction

During the recent A-Sector testing the AØ and BØ compressor buildings were operational. Each building has four Mycom two stage screw compressors. Typically there were four to six compressors operational at a given time. The added compressor capacity was used for transfer line return gas and higher refrigeration capacity. Higher refrigerator capacity was used to minimize power testing interuptions due to marginal refrigeration.

Prior to the A-Sector testing we had a sum total of 20,000 hours of operating time on the eight compressors. During the recent testing, we experienced an additional 20,000 hours of compressor operation.

The BØ compressor system was brought on line 12-16-81 for system decontamination. The system was under microprocessor (μp) based control for the first time. This gave us the capabilities of:

a. Remote compressor starting and stopping

b. Automatic capacity regulation

c. Remote capacity regulation

d. Remote analog and digital readback

e. Remote alarms

Since BØ is the interface point between the helium storage and the rings, the μp controlled inventory makeup or kickback gas as well.

The AØ compressor building was brought on line under μp based control later in the run (4-6-82). This added compressor capacity allowed us to do maintenance on BØ compressors without disturbing the A-Sector testing.

Compressor Control Strategy

Each compressor building has nine control loops (BØ has two additional loops for inventory control). Eight loops control the high and low stage sliders for the four compressors. The ninth loop controls an overpressure clipping valve (EVHP) which diverts gas back to suction.

The two slider loops per compressor controls the compressor throughput and efficiency. Compressor throughput is controlled by the low stage slider. This slider is servoed to maintain a constant discharge pressure.

The high stage slider regulates the compressor interstage pressure. Regulation of the interstage pressure minimizes compressor power consumption and increases rotor bearing life.

Figure 1 schematically shows the high and low stage regulation loops during normal operation (discharge pressure regulation). During certain operating conditions, the compressor building can become isolated from the helium makeup flow located at BØ. Under this condition, the compressors will operate in a suction pressure regulation mode (Figure 2). The low stage slider now regulates the suction pressure to assure that the compressors do not pull a vacuum.

During the A-Sector testing, we ran only in the discharge pressure regulation mode. High stage slider loops were set to regulate the interstage pressure at 20 psig. Low stage slider loops were set to regulate the discharge pressure at "cascading" levels. The eight compressors were made to regulate at different discharge pressures to reduce oscillations/crosstalk between compressors. Set points for the eight compressors incremented by 1 psi from 278 psi to 285 psi. Clipping of the high pressure during transients was accomplished by EVHP at AØ and BØ with set points of 286 and 287 psi respectively.

Helium gas inventory control was accomplished using two μp controlled valves at BØ. In the event of a net helium demand in the ring, EVLP throttles high pressure helium from CHL storage to the ring compressor suction header. The servo loop was set to maintain a 1.0 psig suction pressure. A pressure regulator in parallel acted as a mechanical backup with a set point of 0.5 psig.

During times of net excess helium in the ring, gas is sent back to CHL through EVKI. This loop kickbacks gas when the suction pressure exceeds 1.5 psig. Gas is being kicked back under normal operating conditions since the ring is drawing LHe from an external source (CHL).

Compressor System Problems

During the course of the A-Sector run, we encountered several electrical and mechanical failures. The most serious problem was the loss of power to compressor buildings. On one occasion, feeder 46 went down, taking both AØ and BØ down. This type of failure causes a quick depressurization of the high pressure. As a result, oil pumps cavitated when attempting to start up since it was trying to pump an oil/helium foam mixture. A-Sector testing halted for two days due to the failure.

On two occasions, the BØ substation tripped on overcurrent. Subsequent current measurements showed that the trip setting was too low. Following the second trip, the overcurrent trip and voltage tap settings were increased.

During the course of the A-Sector run we experienced several Cutler Hammer starter failures which took down individual compressors. Failures included two overcurrent modules, a mainframe phasing card, and an electronic coil assembly. The later device did not fail safe, latching the compressor in the ON position. These failures prompted communication with Cutler Hammer which resulted in our receiving detailed manuals on the starter components.

Several control loop related electrical failures also affected individual compressor operation. Three solenoids for the compressor slider actuators burned out during the run. Each time a solenoid burned out, the solid state relay which controls it also burned out. We increased the cycle time for the control loops in order to reduce the repetition rate for firing the solenoids. This cured the problem without adversely effecting the control loops ability to regulate.

Other electrical problems included a actuator position LVDT failure, a 24 volt Lambda power supply failure, and four microswitches for the high stage slider circuit going out of adjustment due to vibration. The later problem resulted in the inability of starting the compressor. Both the high and low stage microswitches must indicate that the stages are fully unloaded before starting.

Mechanical problems observed during the A-Sector run included; a valve actuator drive belt failure, loosening of a Balston oil filter element, a oil pump coupling failure, and inventory control valve problems. The control valve problems were twofold; firstly the control valve size (C_v) was too large initially for the A-Sector run, secondly the force required to move the valve was at the limit of the actuator ability. Higher force actuators are being purchased to correct this problem.

Future Operation

For the E-F Sector testing and subsequent ring operation, several additions need to be made to the compressor hardware and controls software. Hardware changes include:

a. A new 3" high pressure gas line in parallel with our present makeup/kickback line. The present line will be a clean gas supply/return line while the new line will be used for purifying and dirty gas return. (see Figure 3).

b. New inventory control valves sized to the expected load.

c. Environmental control of the compressor controls building. Operating temperature range for the link is 55°F to 90°F.

Compressor software changes include:

a. Updated compressor pages (pages 73,76)

b. Digital status page

c. Global Compressor page

d. Resident local intelligence (RLI) for compressors

e. Capacity control software to limit the number of compressors which regulate. No more than one compressor per building will be allowed to regulate, others being either fully loaded or off. Limiting the number of compressors should reduce crosstalk between compressors as per M. I. Martin proposal.

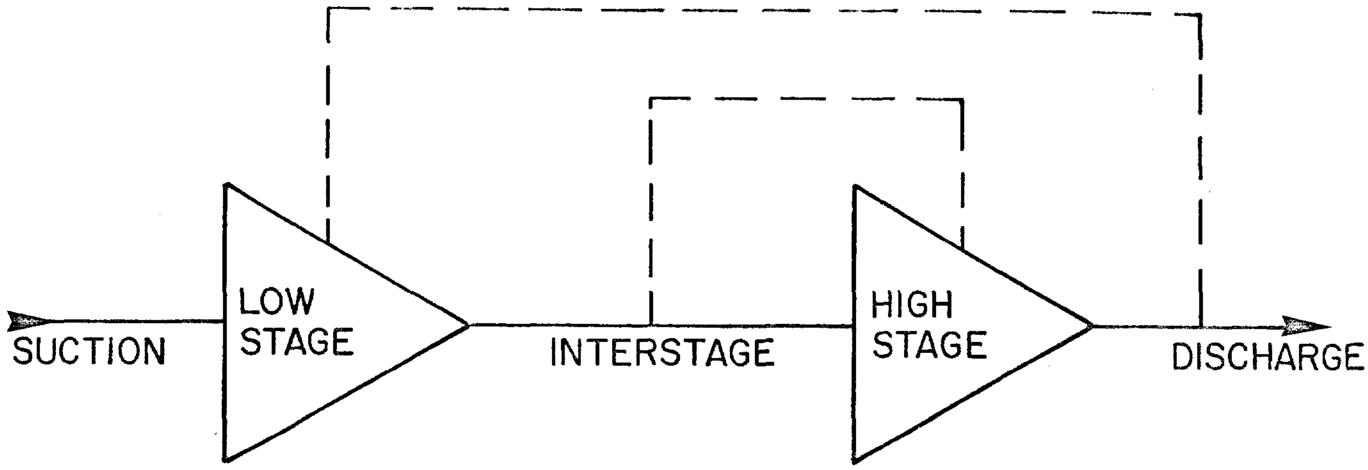
f. Suction regulation mode capabilities

g. Excess capacity software to assure that one to two excess compressors are operating to help absorb system oscillations and quenches

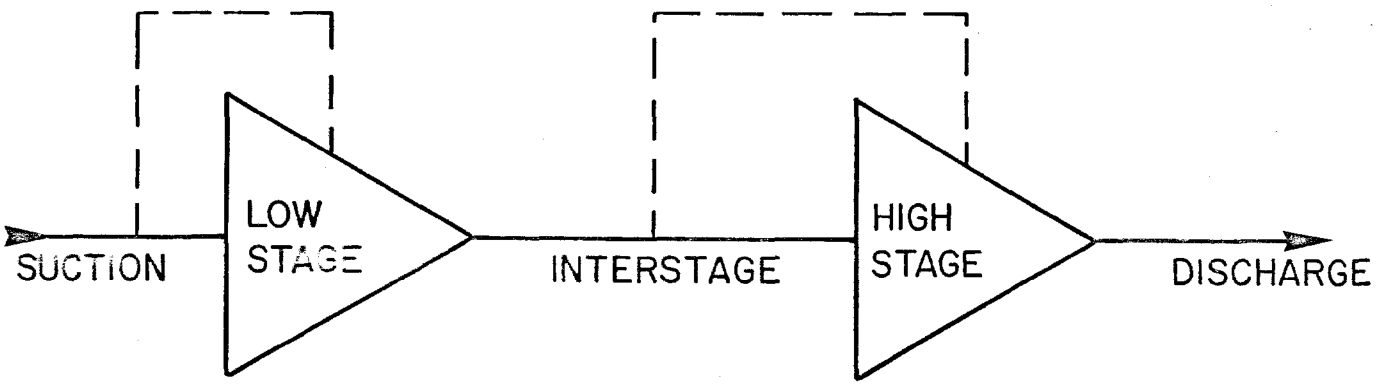
h. Fast compressor unloading to respond to a refrigerator feeder failure

Conclusions

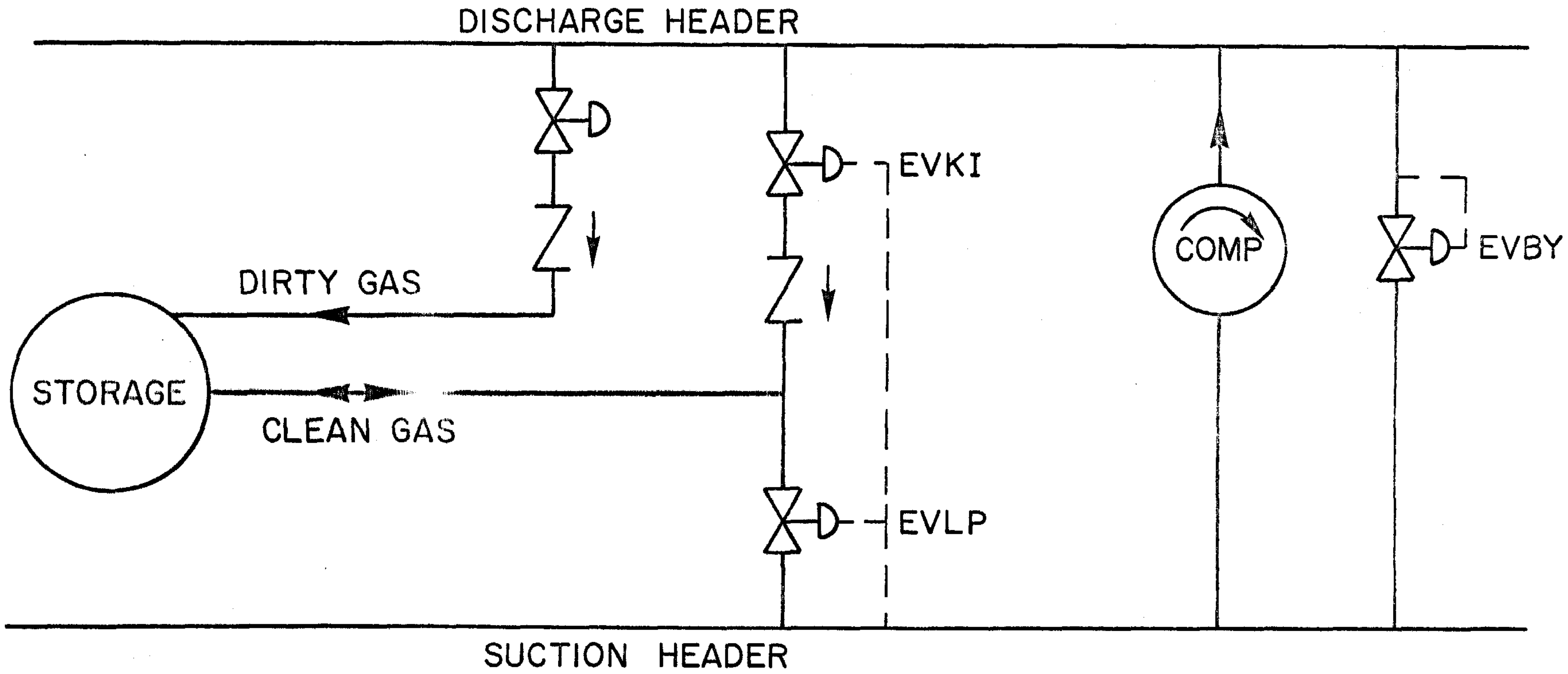
The compressors have been a reliable element within the system. No major maintenance has been necessary to date. Items outlined in the previous section are presently being initiated for future running, with most being ready for the E and F Sector run.



DISCHARGE REGULATION MODE
FIGURE 1



SUCTION REGULATION MODE
FIGURE 2



BO CONTROL VALVE SCHEMATIC

FIGURE 3

Fermilab

VI. EXPANSION ENGINE OPERATION

T. Peterson

Introduction

This summary of expander operations includes refrigerators at A1, A2, A3, and B1 from January, 1982, through June, 1982. The wet engine and dry engine in each of these refrigerators is instrumented with inlet and exhaust pressure gauges, inlet and exhaust vapor pressure thermometers with gauges, a cylinder pressure transducer, and an hour meter. Engine speed is controlled by a DC drive which has a tachometer with a digital speed readout and a "power out" meter which indicates the electrical power generated or used by the system.

Data were taken from these gauges and meters by technicians on operations, by technicians working on engine maintenance and assemblies, and less frequently by staff. During the course of operations in monitoring engine performance, samples of these data were used in estimating efficiencies when temperatures were within the VPT ranges. These data were also used in comparing the horsepower readout with that calculated from conditions.

Although each refrigerator has two kinds of engines, a "wet" engine and a "dry" engine, the use of two kinds of dry engines, a "Gardner-Fermi" dry and a "CTI" dry, means that the mechanical and thermodynamic performances of three kinds of engines are summarized below.

Mechanical Performance

Table 1 summarizes dry engine operations for the sector test, Table 2 wet engines, and Table 3, the problems and proposed solutions. The dry engines clearly each have an area which must be improved: the piston shaft seal on the Gardner-Fermi and the driveshaft on the CTI. A redesign on each of these will be ready for test in July.

The sector test also clearly demonstrated the need for a well-organized engine monitoring and maintenance program. It is necessary to make regularly scheduled checks and log and review the data several times per week, clean and grease the engines and make minor adjustments every 250 hours, do major maintenance and overhauls, keep maintenance records and schedules, order and stock parts and tools, etc. Preliminary estimates are that a crew of at least seven full-time people would be required to take care of the full ring of engines.

Increasing the intervals between major maintenance will, of course, be necessary to decrease accelerator down-time, but since major maintenance is less than a third of the engine work, even with 6000 hours between overhauls the crew will still require at least five full-time people. This estimate does not include analysis of failures, continuing design work on improvements, testing, and other engineering activities.

Efficiencies

Engine efficiency is here defined as the real change in helium enthalpy estimated from inlet and exhaust temperatures and pressures divided by the change in enthalpy which would result from isentropic expansion from the measured inlet pressure to the measured exhaust pressure. The lowest and highest efficiencies are shown in Tables 1 and 2. No trend of efficiency with time was observed in any engine.

The lower wet engine efficiencies sometimes occurred isolated between consistently high readings and may have been due to an inaccurate or non-steady state reading. Sometimes, a consistently low series of efficiencies spontaneously improved or improved following a warmup, indicating contamination such as frozen nitrogen. Generally wet engines efficiencies were above 0.70.

The cause of low CTI dry engine efficiency at A2 is not definitely known, but slightly damaged or dirty valve seats are suspected.

The Gardner-Fermi dry engines at A1 and A3 with low efficiencies in early May were found to have leaking tie-down seals, which allowed high-pressure helium to leak through to the low-pressure side bypassing the engine and then mixing with its exhaust gas resulting in lower net efficiency.

Mass Flow Rate and Horsepower

The refrigeration provided by an expander depends not only on its efficiency but also on the mass flow through the engine. Cylinder pressure traces with corresponding data were analyzed for all three kinds of engines in order to estimate gas density in the cylinder and mass flow. Figure 1 shows the results for the Gardner-Fermi dry engine at B1.

Pressure drops into the engine cylinder as measured on pressure traces were used to estimate e_v, an inlet friction-loss factor (cf. Bird, Stewart, and Lightfoot, Transport Phenomena, ch. 7). e_v is assumed to be a function of the inlet flow geometry alone. For constant density (ρ), the gas velocity (v) is assumed to be related to the inlet presssure drops (Δp) by $\Delta p = (1/2)\ \rho v^2\ e_v$. (Actually, the equations used were more complicated since density was not assumed constant).

For a valve or piping partially blocked by contamination or a valve which fails to open completely, e_v will be greater than normal. The two dashed curves in Figure 1 are examples of the predicted mass flow for an engine with a restricted inlet.

Since mass flow times enthalpy change is the rate of energy removal from the gas, the mass flow can be used with the corresponding temperatures and pressures to estimate horsepower. Figure 2 shows these results for the same engine at Bl. Since the estimated and measured horsepowers agree quite closely over a large range of conditions (see Table 4), the mass flow estimates probably also are accurate to $\pm$ 10 % or better. Figure 3 shows the predicted deviation of horsepower from linearity for the Gardner-Fermi engine for various values of e_v.

A similar estimate of CTI dry engine mass flow and horsepower has been developed from sector test data. The CTI dry engine mass flow and horsepower curves are similar to those for the Gardner-Fermi, but level off more at high speeds due to a greater pressure drop through the intake valve.

The CTI wet engine horsepower estimates are complicated by not having use of the ideal gas relationships. Decrease of horsepower with increasing speed under certain conditions has been observed. In the studies of CTI-W-13 at Bl, the reduction of inlet pressure, the inlet pressure drop to the cylinder, and the recompression due to the exhaust valve resistance to flow as measured on the pressure trace account for the observed roll-off of horsepower with increasing speed.

Nevertheless, the wet engine mass flow estimates (Fig. 4) are probably quite accurate since near and below 6°K the inlet helium density is not a strong function of pressure, so mass flow becomes almost linear with speed.

These mass flow estimates may not apply to a low efficiency engine. For instance, where an engine has a leaking inlet valve or tie-down seal, the mass flow is underestimated by these curves. The horse-powers of the Gardner-Fermi engines with bad tie-down seals were higher than the predicted values. Conversely, if inlet flow is restricted the horsepowers and mass flows may be lower than expected.

Conclusions

1. Both kinds of dry engines need improved mechanical reliability. Designs are complete, and new parts will soon have to be tested.

2. A well-organized maintenance program and a crew of at least five, probably more, full-time people will be required to care for the ring of engines.

3. All three types of engines are capable of excellent efficiencies for as long as they have been run so far. Some specific causes of low efficiency are understood.

4. Mass flow versus speed can be accurately predicted for efficient engines of all three types.

5. The general thermodynamic performance of the engines as reflected in the horsepower is quantitatively understood for dry engines and for some of the wet engine data, including an example of horsepower roll-off with speed.

Table 1

1982 Dry Engine Operations - Sector Test

Location	Engine	Range of Efficiencies	Hours of Operation	Dates Operational	Reason Engine Shut Off
A1	GFD-3	0.52-0.63	1321	17 Aug 80-25 Feb 82	Frost around piston shaft seal
	GFD-7	0.50-0.61	1088	25 Feb 82-19 Apr 82	Piston shaft seal nut came loose
		0.46-0.65	433	19 Apr 82-11 May 82	Frost around piston shaft seal. Bad tie-down seal
		0.60-0.70	207	11 May 82-11 June 82	Piston shaft seal nut came loose
A2	GFD-4	0.57-0.70	900	17 Sep 80-02 Feb 82	Broken impact block. (Also frost problem around piston shaft
	CTID-5	0.40-0.61	641	02 Feb 82-19 Mar 82	Drive shaft failure
	CTID-7	0.46-0.72	807	19 Mar 82-Present	(Replaced DC Motor on 5/4)
A3	GFD-9	0.46-0.68	1800	18 May 81-18 Mar 82	Piston shaft siezure in linear bearing
	GFD-6	0.50-0.58	701	18 Mar 82-11 May 82	Bad tie-down seal
		> 0.65	265	11 May 82-Present	
B1	GFD-2	0.64-0.81	1967	Oct 81 -Present	

Table 2

1982 Wet Engine Operations - Sector Test

Location	CTI Wet Engine	Range of Efficiencies	House of Operation	Dates Operational	Reason Engine Shut Off
A1	2		330	16 Jan 82- 30 Jan 80	Leaking Shut-off valve bellows. Poor performance
	16	0.60 - 0.82	2170	30 Jan 82- 28 May 82	Broken piston O-ring, frosted crosshead, lost cryostat vacuum
		0.74 - 0.77	346	28 May 82- Present	
A2	9	0.77 - 0.82	300	Spring 81- 29 Jan 82	Bad blowby and dry and damaged crosshead on #2 side
			381	29 Jan 82- 15 Feb 82	Bad blowby and dry and damaged crosshead on #2 side
	15	0.65 - 0.88	2104	15 Feb 82- Present	
A3	8		1800	18 May 81- 25 Feb 82	Loose intake valve seat
	11	0.60 - 0.85	2009	25 Feb 82- Present	
B1	13	0.78 - 0.84	2082	Present	

21 June 1982

Table 3

1982 Engine Operations Summary - Sector Test

Engine	Mean time between major maintenance. (Hours)	Primary Problem	Other Problems	Proposed Solutions
Gardner-Fermi Dry	1000	Piston shaft seal lifetime	Shaft seal nut, impact block, tie-down seal, tappet wear, alignment of moving parts.	A new shaft seal is designed - test it. Realign warm ends. New tappets are designed - test them. Tie-down seal and impact block solutions are tested OK.
CTI Dry	700	Driveshaft fails by fatigue near its point of attachment to the crank.	Occasional low efficiency without an obvious cause, bearing and cross-head lubrication and lifetimes.	A new driveshaft - crank assembly is due on hand July 5 - test it. Clean valves and valve seals upon receipt from vendor and keep them clean. (See also wet engine.)
CTI Wet	1800^{+}	Cross head and O-ring lubrication and wear.	Speed regulation, performance "roll-off" with speed, bearing lubrication and lifetimes.	Keeping engines clean and scheduled frequent checks and greasing probably contributed to the improvement after February. Test controllers' speed reg. Monitor bearing lifetimes. See text for "roll-off".

Figure 1 23 June 82

Theoretical Mass Flow vs. RPM
for Gardner - Fermi Dry Engines

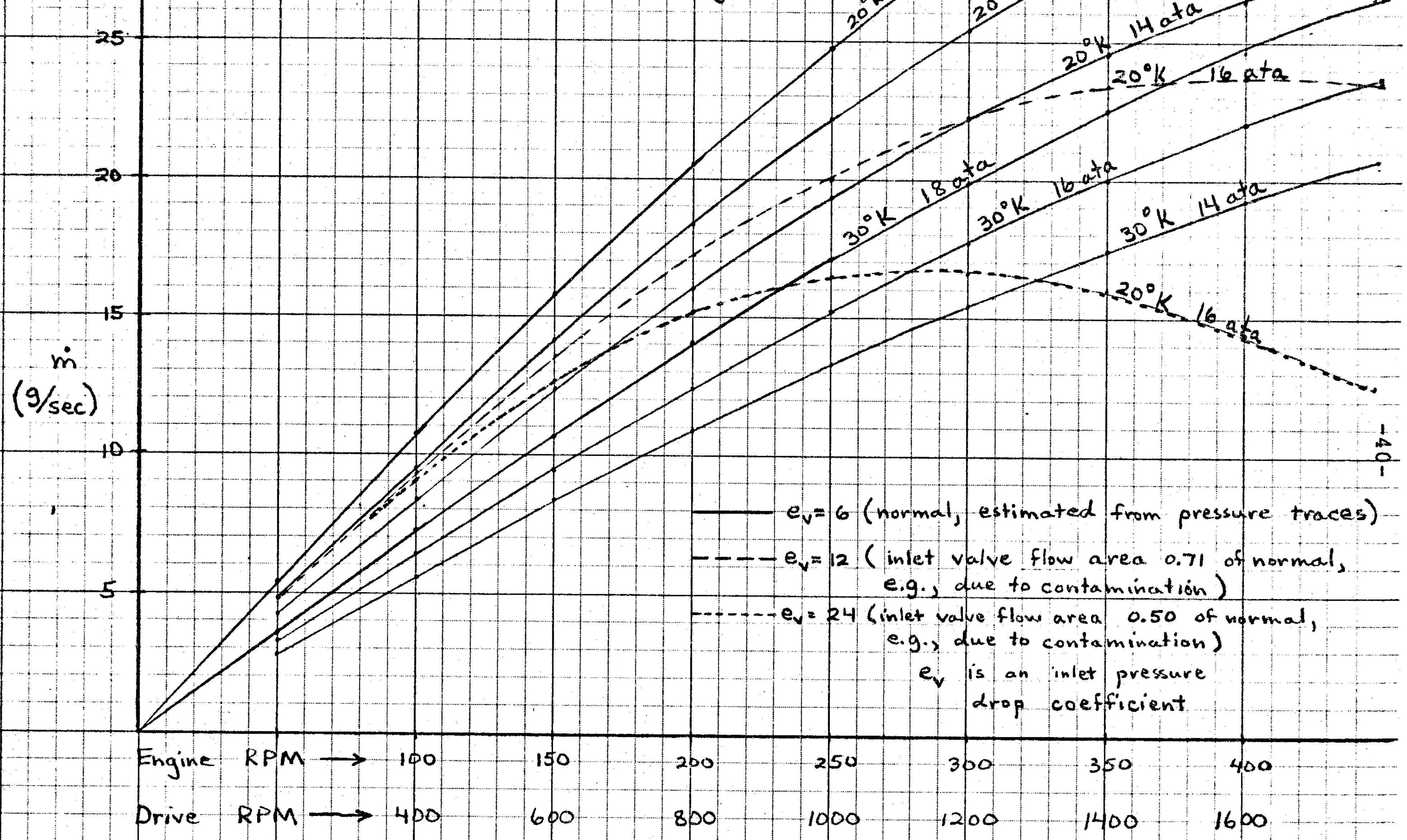
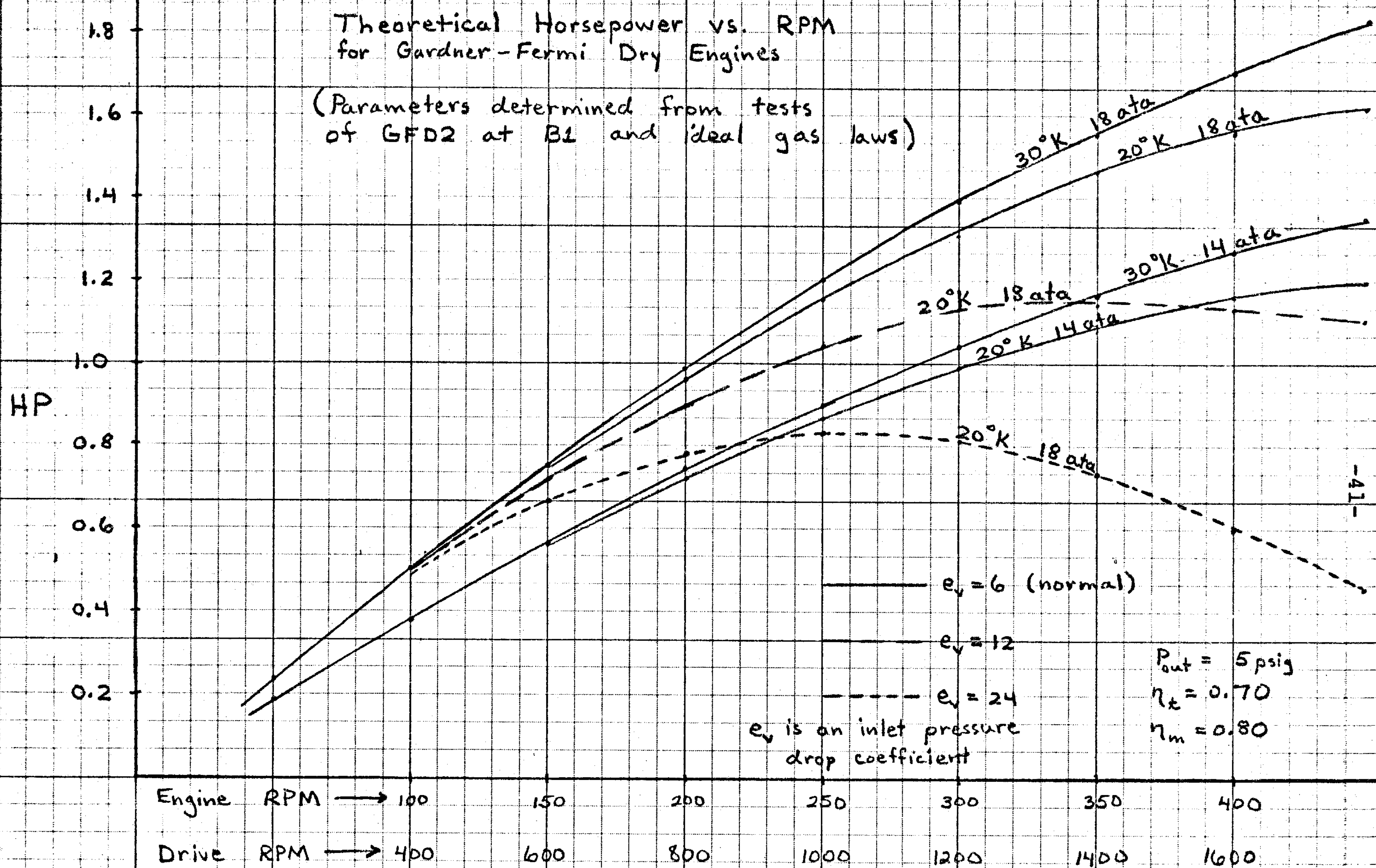


Figure 2

23 June 82



24 June 82

Table 4

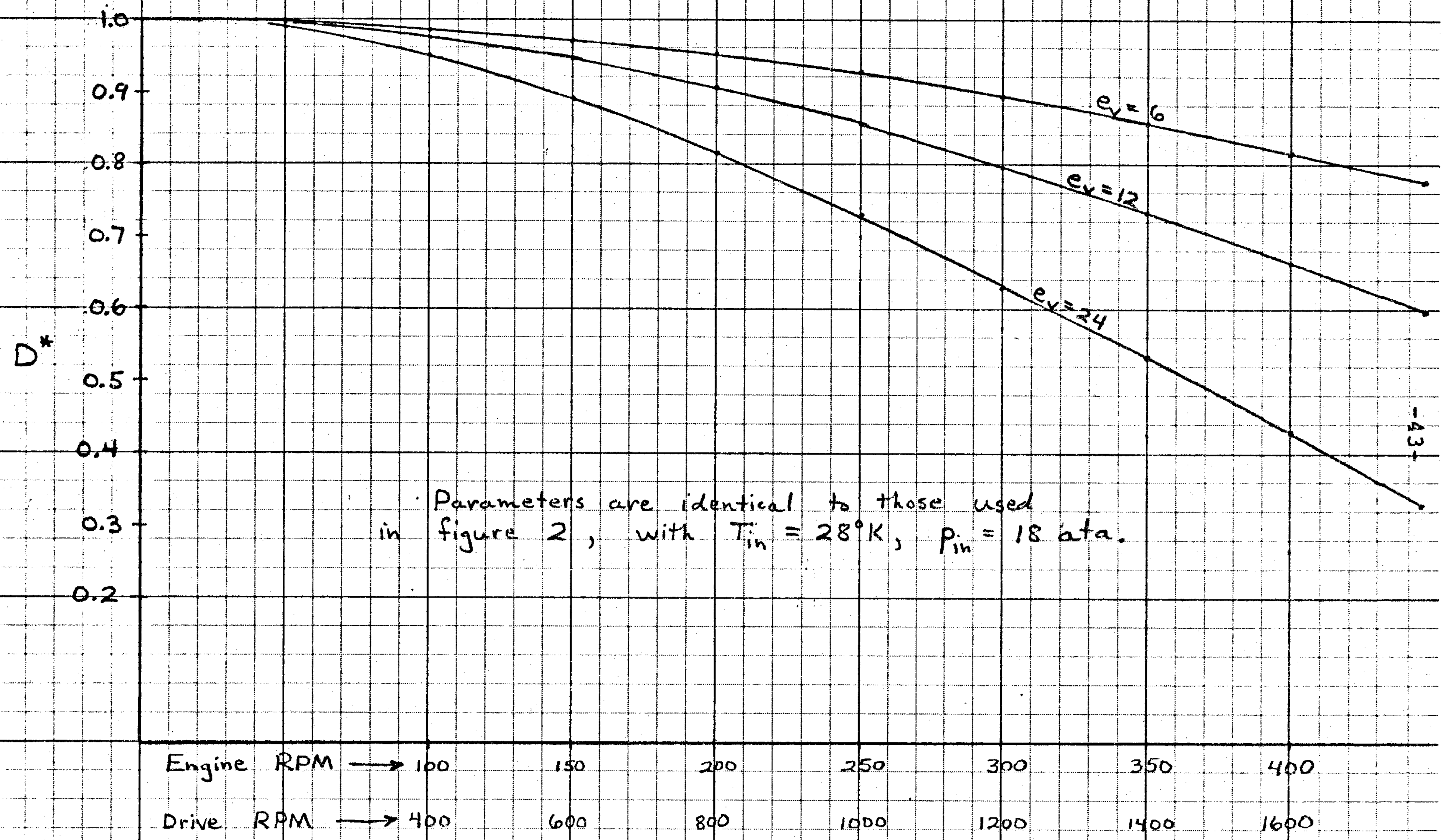
Accuracy of Horsepower Curves in Figure 2

The parameters for the equations from which figures 1 and 2 were made came from refrigeration studies at B1 on May 20, 1982. Those equations were then used to calculate horsepowers for data taken from April 4 to June 3, 1982, and compare that prediction with the meter at B1 with the following results.

Time period-	2 months (1000 hours of operation)
Drive speed range-	361 RPM to 1690 RPM
Inlet pressure range-	215 psig to 270 psig
Inlet temperature range-	$25^{O}K$ to $35^{O}K$
Efficiency range-	0.64 to 0.81
Number of data points-	20
Maximum error in calculating meter horsepower for 20 points-	12%
Average error in calculating meter horsepower for 20 points-	3%

Figure 3

Deviation of Horsepower from Linearity



* D = (Theoretical horsepower like in figure 2) / (Ordinate of line tangent to horsepower curve at origin)

Tom Peterson
3 Feb 82

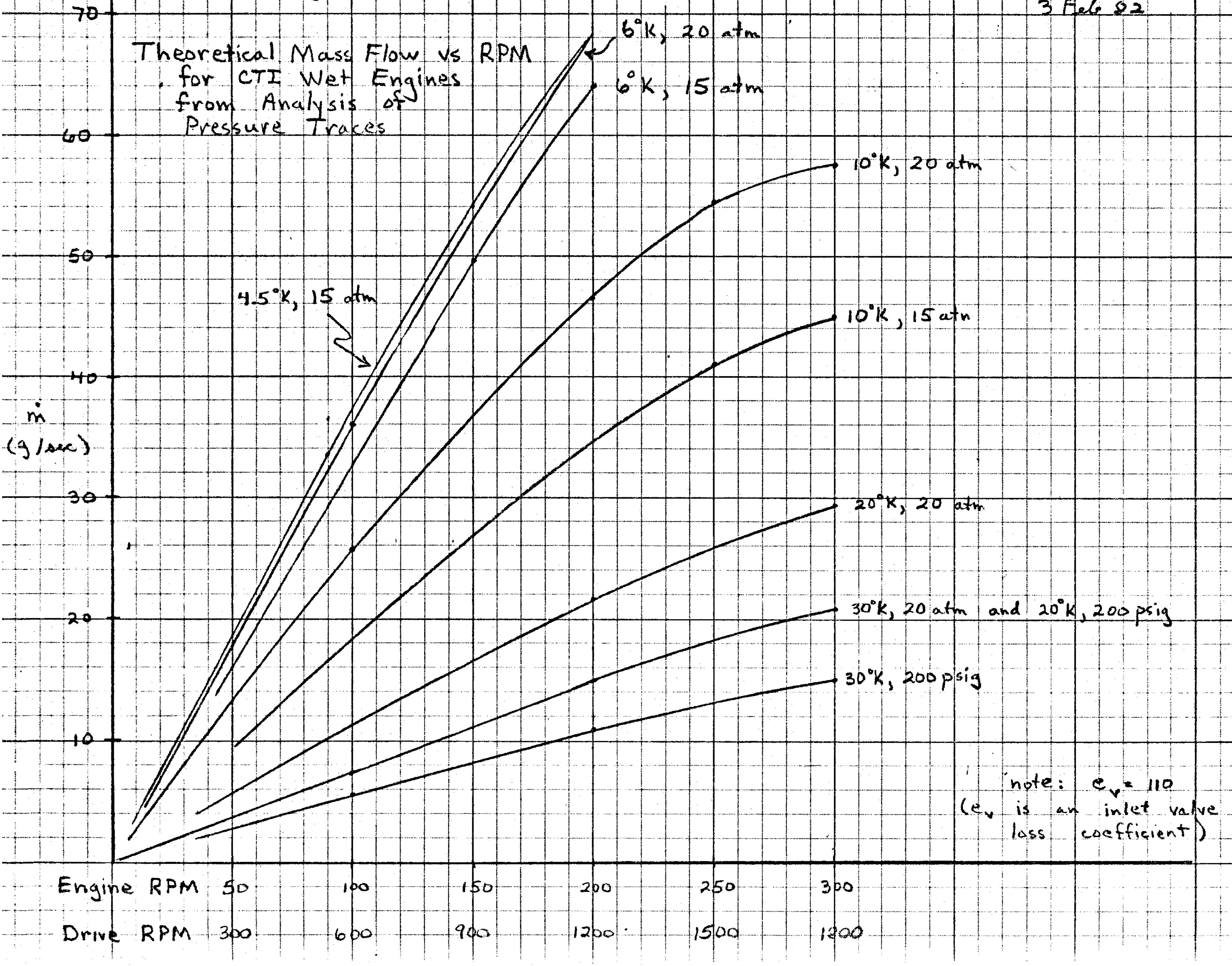


Figure 4

Theoretical Mass Flow vs RPM for CTI Wet Engines from Analysis of Pressure Traces

Fermilab

VII. EXPANSION ENGINE CONTROLLERS

J. Dinkel

GENERAL

For an expansion engine to achieve a specified flow of gas or liquid for cooling, its speed of rotation must be controlled by an external source. The expansion engine drive is a speed regulated dc motor which acts as either a source of power or a load to its associated engine. Motor speed is adjustable from 200 to 1800 rpm. The engine controller incorporates safety interlocks to shut the engine down in case of a malfunction or personnel safety hazard. It also provides local and remote readouts for motor speed and power as well as status data. The basic drives are of two types: for the Dry Engine, a 7.5 hp motor is regulated by a General Electric 6V7F3229 Speed Variator; for the Wet Engine, a 2 hp motor is regulated by a General Electric 6VHR Statotrol regenerative drive.

OPERATIONAL PERFORMANCE

The Dry Engine drive gave excellent performance. Once the factory set-up procedure was learned and the current limit setting reduced, the fuse blowing problem went away.

The Wet Engine drive was a different story. As purchased from GE, the motor controller is an "economy" unit having a fairly low gain and no speed feedback. As a result, the Wet Engine speed would change sometimes as much as 10-15%. It required the better part of this run to pinpoint the cause of the problem and arrive at a solution.

Prior to shutdown, a circuit card was added to the Wet Engine controller at A1 which provided both tachometer feedback and the controlled start and stop feature which was found desirable on the Dry Engine controller. The effect of this modification was to reduce speed variations which had been running at the 10% level to within 1% for the remainder of the run.

During the abnormally cold weather, the SCR's would not reliably fire until the Wet Engine controller was warmed up.

Other failures of the expansion drives were random and minor.

In several cases, the power reversal feature tripped the drive off when an engine malfunction occurred. This should have kept damage to the affected engine to a minimum.

We also observed that the motors are often required to operate at low speed under heavy load. Under these conditions, the internal cooling fans are ineffective and the motor overheats.

Significant ($\sim$5%) differences in absolute calibration (volts/rpm) exist for tachometers mounted on the dc motors. When an engine is changed, the motor goes with it. This means that the drive must be recalibrated each time an engine is changed.

CONCLUSIONS

To ease the task of rewiring the motor and terminal box connections each time an engine is changed, connectors will be added for both the motor connections and the control wiring.

All Dry Engines, with the exception of A-Sector, will be set-up according to factory procedures (see Appendix) with a reduced current limit.

External cooling fans available as an option from GE should be added to the motors.

The modification prototyped in A1 will be added to all Wet Engine drives in the ring so they will have approximately the same performance as the Dry Engine drive.

Appendix I
Description of Existing 7.5 HP Dry Engine Drive

General

The 7.5 HP Dry Engine drive powers a DC motor such that rotation is possible in either direction. In addition, the drive enables the motor to act as either a source of power or a load to regulate its associated engine speed to some given value. Motor speed is adjustable from 200 to 1800 RPM. Its primary function in the satellite refrigeration system is to transfer energy from its associated expansion engine to the 480 volt power line. In addition, it provides the safety interlocks, monitor and control interface for the engine.

Motor Drive

The basic elements of the half wave dc SCR drive are shown in simplified form in figure 1. Three phase ac power is coupled to the control unit through a separately mounted isolation transformer. Input power then goes through fuses, the main contactor to the power conversion module (SCR's) where it is converted to an adjustable dc voltage. DC current is fed through a shunt to the DC motor armature. The return side of the DC motor is connected to the isolation transformer neutral. The speed of the motor is proportional to the dc voltage applied to its armature. Speed is measured by a tachometer directly connected to the motor.

The remainder of the motor control circuitry is situated on 5 printed circuit cards; the power supply card, the main control card, the interface card, the motor field card, and the diagnostic card. Power for the control circuitry is taken from the three phase input through control fuses to the control power transformer. This transformer is fitted with two isolated secondary windings: 115 VAC to operate the main contactor, auxiliary power supplies, and the gas valve solenoid; and 50 VAC tapped to the input of the power supply card.

The power supply card rectifies the ac input and provides regulated plus and minus 20 volts for the control circuits. Unregulated plus and minus 30 volts dc is also provided to drive the static logic switches. All of the dc outputs are fused to protect against overloads. The plus and minus 20V dc outputs are protected against over voltage conditions caused by a power supply failure.

The primary purpose of the main control card is to drive the power conversion module as directed by the speed reference and feedback signals. This card also performs several additional functions such as linear timing of the reference; current or torque limit; "Ready to Run" indicator; and various scaling and trimming adjustments. The output of the linear timed reference is compared to a voltage derived from the tachometer (SFB) in the error amplifier. The result will be a low voltage (near zero) when the drive is at regulating speed. The result, however, will not be low when the drive is in either current limit or CEMF limit. To maintain good load regulation, the error amplifier output is fed into the regulator integrator. The output of the integrator is the reference to the SCR driver.

To protect the system, two limit circuits are provided, counter-EMF or CEMF limit and current or torque limit. The output of these limit circuits also drive the regulator integrator and will override the error amplifier if required. The current limit is set with the CUR LIMIT adjustment and the counter-EMF limit is set with the CEMF LIMIT adjustment. Typically the current limit is set to the motor name plate which corresponds to 2.5 volts of current feedback CFB; the counter EMF is normally limited to 250 armature volts at no load or 5.75 volts of CEMF. The counter-EMF signal CEMF is developed by subtracting a voltage proportional to the IR drop of the motor from the armature voltage feedback.

The SCR driver reference, the armature voltage feedback, VFB, and the damping adjustment DAMP are summed at the input to the driver. The driver converts this error to pulse trains which drive the SCR gates in such a maner as to maintain the motor voltage proportional to the driver reference. The damping adjustment DAMP controls the response of the driver. Generally speaking, DAMP is used only to quiet small oscillations which occur in the current under light load conditions. Too much damping will slow down the system response and tend to cause excessive overshoot.

The interface card has several purposes: to provide low level isolated signals corresponding to the three phase ac, dc armature voltage, armature current, and tachometer feedback; to control the start, stop and synchronizing of the drive while monitoring the system for abnormal operating conditions.

The motor field exciter card provides a fixed value of field excitation. The field excitation is reduced by 50 percent when the main contactor is open. A field loss circuit is provided to shut down the drive upon loss of field excitation.

The diagnostic card performs no function under normal operating conditions but will program the drive into a diagnostic run mode and diagnostic static mode for ease in start up and troubleshooting.

Monitoring and Control

Two local readouts are provided on a local control panel. Motor speed is given as a 4 digit display. Its input is derived from the speed feedback (SFB) voltage in the drive unit. A remote readout is provided with a sensitivity of 1 volt/200 RPM. Drive power is provided on a similar display. The horsepower reading is generated by multiplying a measure of the armature current CFB and the armature voltage VFB. A remote readout is provided with a sensitivity of 1 V/H.P. A motor revolution counter is provided to log engine performance. Status of the protective trips is also displayed. Five trips are presently in service. Emergency Brake indicates that the overspeed brake on the engine has tripped. Power Reversal indicates that the motor has been supplying drive torque to the engine over a period of time. The drive torque is related to the motor armature current by the factor .85 ft. lb./amp. The armature current feedback CFB is therefore a measure of the drive torque by the relation: PR=4.74 volts/ft. lb. It comes from an amplifier which has a 20 second time constant to prevent erroneous trips. When the voltage PR exceeds some preset value, a power reversal condition is said to exist. Motor Overtemp indicates an overtemperature condition in the drive motor. Motor Overspeed indicates that the speed of the motor as measured by SFB has exceeded some preset value. Emergency Stop indicates that someone has pushed the Emergency Stop push button on the engine. These trips may be read remotely and may be reset by a local push button or remotely.

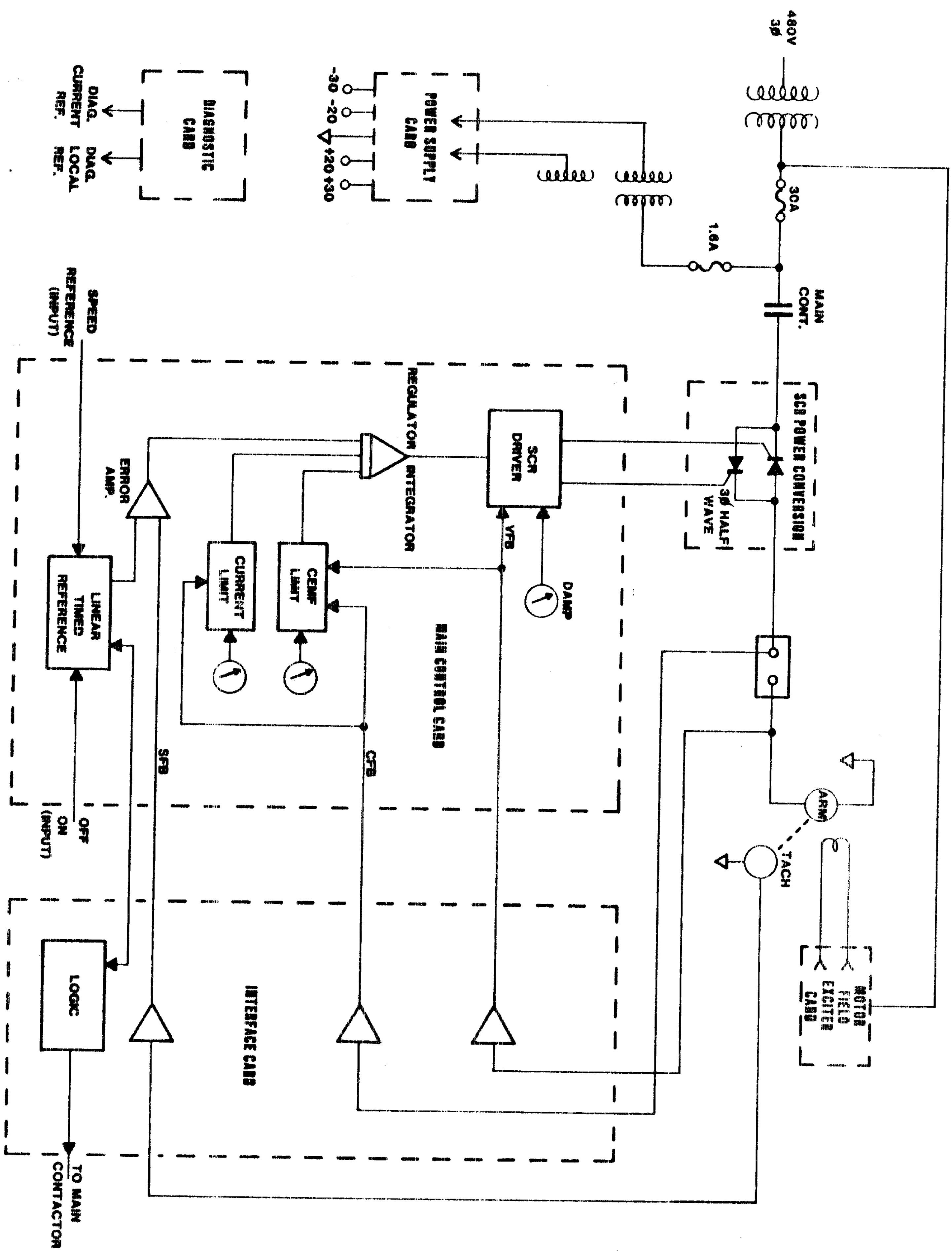
Motor speed may be set locally with a 10-turn control. A circuit is provided to limit the motor speed range between a minimum of 200 RPM and a maximum of 1900 RPM. Both minimum and maximum points may be adjusted. A Local/Remote switch is provided to select either a local or remote speed reference and ON command.

The timed reference in this drive ramps the speed at a constant preset rate from one setting to another. When the ON button is pushed, the timed reference closes the contactor and ramps the engine up to speed. When the OFF button is pushed, the timed reference ramps the speed to zero. At some preset minimum speed, the contactor is opened.

CALIBRATION

1. Turn the power on but leave the motor off.
2. Adjust H.P.ZERO on MONITOR card for a zero reading on the H.P. display.
3. Remove the yellow wire going to CFB and the orange wire going to VFB. Using an external power supply, apply +2.00 Vdc to both wires referenced to ground. Adjust H.P. CAL for a reading of 2.62 V at U3 pin 7.
4. Turn COMP full C.C.W.
5. Set GAIN to 9 o'clock positon (min).
6. Set RESP to 9 o'clock position (min).
7. Turn DAMP full C.C.W.
8. Set LINTIME to 9 o'clock position.
9. Monitoring U2, pin 7, adjust INT. ZERO on MONITOR card 1 for a zero or slightly negative (.1-.2V) reading.
10. Set FLOSS just to the point where the READY TO RUN l.e.d. stays on when the RESET button is pressed.
11. With SPEED REFERENCE set to the middle of its range, turn motor on.
12. Meter U4, pin 3 and adjust OS TRIP on MONITOR card for desired overspeed trip voltage according to the factor 200 rpm/volt (≈9V).
13. Meter U1, pin 3 and adjust E.T. on MONITOR card for a reading of 6.00 V.
14. Meter terminal #28 on terminal strip 2TB and adjust SPEED REFERENCE for a reading of 5.00 V.
15. Adjust REF SCALE for a reading of 1000 rpm on the external tachometer.

16. Adjust SPEED CAL on MONITOR card to give a reading of 1000 on the RPM display.
17. Adjust RPM CAL for 5.00 V on remote control connector J1-B.
18. Set SPEED REFERENCE to its maximum (full C.W.). Adjust MAX SPEED on INTERLOCK card for maximum speed.
19. Meter DA1 (-) and XO (+) and adjust COUNTER EMF on MAIN board for a reading of 250 V.
20. Adjust MAX SPEED on MAIN board for a reading of 1800 rpm on the external tachometer.
21. Meter test point 1LA and adjust CUR LIMIT for a reading of 2.00 V.
22. Set SPEED REFERENCE to its minimum position (full C.C.W.) and adjust MIN SPEED on INTERLOCK card for a reading of 200 rpm on external tachometer.


480V 3ø
30A
1.6A
MAIN CONT.
SCR POWER CONVERSION
3ø HALF WAVE
POWER SUPPLY CARD
-30 -20 +20 +30
DIAGNOSTIC CARD
DIAG. CURRENT REF.
DIAG. LOCAL REF.
SCR DRIVER
DAMP
VFB
REGULATOR
INTEGRATOR
ERROR AMP.
CURRENT LIMIT
CEMF LIMIT
MAIN CONTROL CARD
LINEAR TIMED REFERENCE
SPEED REFERENCE (INPUT)
OFF ON (INPUT)
SFB
CFB
INTERFACE CARD
LOGIC
TO MAIN CONTACTOR
ARM
TACH
MOTOR FIELD EXCITER CARD

Appendix II
Description of Existing 2 HP West Engine Drive

General

The 2 HP Wet Engine drive powers a DC motor such that rotation is possible in either direction. In addition, the drive enables the motor to act as either a source of power or a load to regulate its associated engine speed to some given value. Motor speed is adjustable from 200 to 1800 RPM. Its primary function in the satellite refrigeration system is to transfer energy from its associated expansion engine to the 480 volt power line. In addition, it provides the safety interlocks, monitor and control interface for the engine.

Motor Drive

Single phase 220 volt AC power is coupled to the control unit through a separately mounted isolation transformer. Input power then goes through fuses, the main contactor, to the SCR's where it is converted to an adjustable DC voltage. DC current is fed through a 0.5 ohm shunt to the motor armature. The speed of the motor is proportional to the DC voltage applied to its armature. Speed is measured by a tachometer directly connected to the motor.

The remainder of the motor control circuitry resides on a single printed circuit card. The primary purpose of this circuitry is to drive the SCR's as directed by the speed reference and feedback signals. The input speed reference is compared to a voltage proportional to the armature voltage. The result is amplified by a factor of 15, filtered, and applied to a phase shift and overlap circuit. This circuit provides the proper timing for the firing circuits. The overlap circuit provides for the transfer from the forward to the reverse firing circuit which in turn causes the motor to act as either a source of power or a load.

Monitoring and Control

Two local readouts are provided on a local control panel mounted in an enclosure directly above the motor control unit. Motor speed is given as a 4 digit display. Its input is derived from the output of the DC tachometer. A remote readout is provided with a sensitivity of 1 volt/200 RPM. Drive power is provided on a similar display. The horsepower reading is generated by multiplying a measure of the armature current as seen by the shunt and the armature voltage. A remote readout is provided with a sensitivity of 1 V/HP. A motor revolution counter is provided to log engine performance. Status of the protective trips is also displayed. Four trips are presently in service. <u>Emergency Brake</u> indicates that the overspeed brake on the engine has tripped. <u>Power Reversal</u> indicates that the motor has been supplying drive torque to the engine over a period of time. The drive torque is related to the motor armature current by the factor .46 FT.LB./AMP. The armature current feedback is therefore a measure of the drive torque by the relation: PR=23.4 volts/ft. lb. It comes from an amplifier which has a 20 second time constant to prevent erroneous trips. When the voltage PR exceeds some preset value, a power reversal condition is said to exist. Motor Overspeed indicates that the speed of the motor as measured by the tachometer has exceeded some preset value. Emergency Stop indicates that someone has pushed the Emergency Stop push button on the engine. These trips may be read remotely and may be reset by a local push button or remotely.

Motor speed may be set locally with a 10-turn control. A circuit is provided to limit the motor speed range between a minimum of 200 RPM and a maximum of 1900 RPM. Both minimum and maximum points may be adjusted. A Local/Remote switch is provided to select either a local or remote speed reference and ON command.

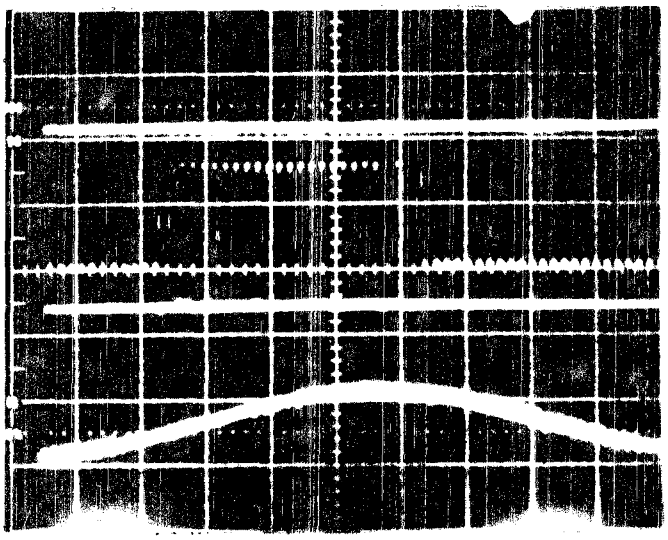
No timed reference exists for this drive. When the ON button is pushed, the motor applies maximum torque to get the engine up to speed. When the OFF button is pushed, the main contactor disconnects the drive from the line and shunts a 7.5 ohm resistor across the armature to provide dynamic breaking to the motor.

Calibration

With power on, but the motor off, adjust HP ZERO on the monitor card to zero the H.P. display. To zero the power reversal trip amplifier, attach a meter or scope to U2-7. Adjust INT ZERO so that the reading is zero or slightly negative (.1-.2 volts). A diode prevents this amplifier from going beyond -.5 volts.

Turn motor on with the input reference set at 5.0 volts. Motor speed should be 1000 RPM as measured with an external tachometer. With motor running at 1000 RPM, adjust SPEED CAL on monitor card to give 1000 on the RPM display. Adjust RPM CAL for a 5.00 volt output on remote control connector pin J1-B. Meter U4-3 and adjust OS TRIP for desired overspeed trip voltage according to the factor 200 RPM/volt. Meter U1-3 and adjust ET for the desired trip level of reverse torque. Turn the speed reference to zero and adjust MIN SPEED for 200 RPM. Set the speed reference to 10 volts. The maximum speed will probably be limited by the maximum available armature voltage (approximately 250 volts) to approximately 1830 RPM. MAX SPEED should be adjusted for a drop of 10-20 RPM.

To check the setting of the OVERLAP adjustment on the motor control circuit board, connect a function generator (Tektronix FG501 or equivalent) between 2TB-B and C. The frequency is set to 100 Hz, and the amplitude is 1 volt p-p with a 1.5 volt positive offset. Monitor firing circuit inputs TP42 and TP44. Adjust OVERLAP to achieve a transfer from one circuit to the other with neither dead zone nor overlap. When properly set up, it should appear as in the photograph of Figure 1.



TP42 - 2 volts/div

TP44 - 2 volts/div

Input Ref. 1 volt/div
Sweep 100 ms/div

Figure 1

VIII. HEAT LOAD MEASUREMENTS

J. C. Theilacker

INTRODUCTION

Several tests have been performed to measure the static refrigeration heat load in the magnets. Towards the end of the A-Sector run, we also measured the combined static and dynamic heat loads at several ramp levels and cycle times. Results from the A-Sector tests are presented in the next section with recommendations for further testing presented in the last section.

Determination of the static heat load for the magnets is an important parameter for understanding the operation of the satellite refrigerators in stand-alone mode. During the beginning of the A-Sector run (January 1982) we had considerable difficulty in filling the magnets (see Section II). We attributed this to a high static heat load. Figure 1, illustrates the problem. The design refrigeration production is 623 watts and the design liquefaction rate is 126 ℓ/hr. For combined refrigeration/liquifection loads the output is approximately linear (shown linear in Figure 1 for illustration purposes). If the static heat load is large, then the reserve capacity left for liquefying is small. As a result, the static heat load determines how fast the satellite will be able to fill the magnets.

Determination of the combined static and AC heat loads is also an important parameter for extrapolating the satellite capacity to various ramps and cycle times. This is useful for determining the limiting factor in the accelerator (refrigeration, power supplies, etc), possible beam loading estimation in areas such as E1 and A1, and refrigerator degradation. (More data points are required in order to make these extrapolations).

The method used to determine the heat load, whether static or combined, is illustrated in Figure 2. Supply helium from the refrigerator is maintained subcooled throughout the test. At the beginning of the test, 2Ø helium returns from each string to the refrigerator. During the test, the flow control valves EVUH and EVDH are throttled until the return gas from each string begins to superheat. The flow control valves are throttled typically, 2 to 3% per hour at the beginning of the test and 0.5 to 1% per hour towards the end.

The object is to have the superheat instrumentation (DT16 and DT19) read a constant 1 to 2 psi of superheat for longer than 30 minutes. One heat load measurement can easily take 6 to 8 hours.

RESULTS

Heat load data from the June 1982 testing was not consistent with previous static heat load data on the same magnets. Previous static heat load measurements on the same magnet strings showed 500 watts per refrigerator, while current testing showed 350-400 watts. The source of this discrepancy has not been found, although possibilities include:

1. Instrumentation errors/problems

   -A2 DT16 is suspected to be undercharged
   -FIUH/DH had been oscillating at the end of the run. When in satellite mode, FI4 (high pressure gas inlet) was used instead.

2. Different care/experience levels in testing operators

3. A long "soaking" time, resulting in non steady-state heat loads to be measured

4. Vacuum variations

All of the dynamic load testing was performed at A2. Although the magnitude of the load was not consistent with previous testing, we feel that the trends observed at various ramp levels and cycles are accurate. Data was taken with the refrigerator in satellite mode. A summary of the results is given in Table 1.

Current (Amps)	Cycle (sec)	$\dot{I}$up (A/S)	$\dot{I}$down (A/S)	Load (Watts)
0	-	-	-	350-400
50-4000	65	170	130	465-515
50-4000	47	259	190	570-610
350-4000	47	259	190	540-590

Combined Static and Dynamic Heat Loads
Table 1

AC losses can be divided into two types, hysteresis and eddy current losses. Hysteresis losses are proportional to the change of current associated with the ramp as shown in equation 1. The data of table 1 can be plotted to show the

$$\text{Hysteresis loss} \propto \frac{\int_0^\tau |\dot{B}|\,dt}{\tau} \propto \frac{\int_0^\tau |\dot{I}|\,dt}{\tau}$$

$$\propto \frac{\dot{I}up \,\frac{\Delta I}{\dot{I}up} + \dot{I}dn \,\frac{\Delta I}{\dot{I}dn}}{\tau}$$

$$\propto \frac{\Delta I}{\tau} \cdot 2 \tag{1}$$

Where $\Delta I = I\,max - I\,min$

total AC loss dependance on this hysteresis term (Figure 3). The nonlinearity of the data suggests that there is a second order loss (i.e., eddy current loss).

Eddy current losses are proportional to the ramp rate, as shown in equation 2. Plotting the Ac losses versus the eddy current loss term results in Figure 4. The linearity of the data in Figure 4 compared to Figure 3 suggests that the eddy current portion of the heat load may be larger than the original 10-20% found in past work. However, error bands in the data make it impossible to conclusively proportion hysteresis and eddy current losses.

$$\text{Eddy Current Loss} \propto \frac{\int_0^\tau \dot{B}^2\,dt}{\tau} \propto \frac{\int_0^\tau \dot{I}^2\,dt}{\tau}$$

$$\propto \frac{\dot{I}^2_{up}\,\frac{\Delta I}{\dot{I}up} + \dot{I}^2 dn\,\frac{\Delta I}{\dot{I}dn}}{\tau}$$

$$\propto \frac{\Delta I}{\tau}\left(|\dot{I}up| + |\dot{I}dn|\right) \tag{2}$$

Recommendations

Further testing is required to reach any concrete statements concerning AC losses. The following test sequence is recommended during the E-F Sector testing.

1. Measure the static heat load in two refrigerators early in the run. Refrigerators should be in satellite mode in order to achieve repetitive flow measurement. A venturi spool piece could also be made to measure the flow from CHL. This device can also be used to calibrate the liquid helium control valve (EVLH).

2. Measure combined static and AC loads with the same two refrigerators at various ramp levels, rates, and cycle times. These parameters should be chosen to give a large spread in $\Delta I/\tau$ and ΔI $(|\dot{I}up| + |\dot{I}dn|)/\tau$ in order to follow treads more closely. (i.e., Heat load wattage spread from static load to 1000 watts)
3. Repeat step 1 later in the run in order to test for a "soaking" effect.

The refrigerators should be chosen on the basis of instrumentation response and reliability.

Keywords

AC Loss
Capacity
Heat Load
Magnet

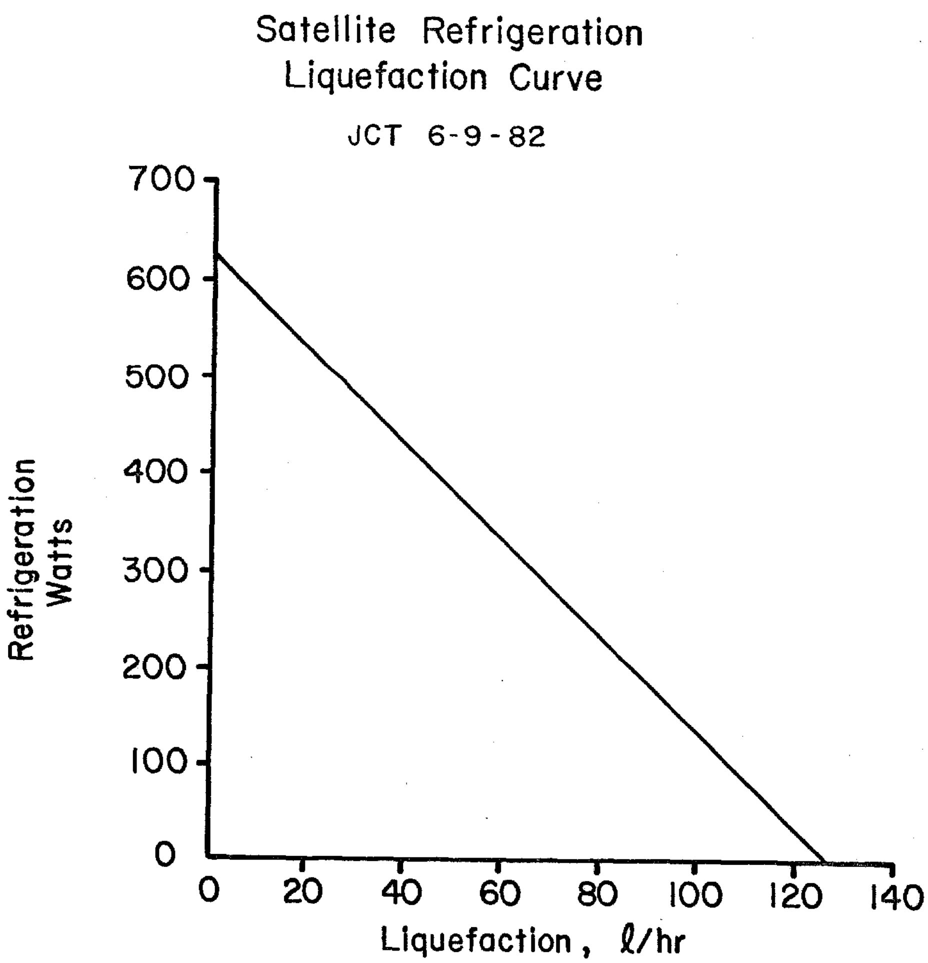

Satellite Refrigeration
Liquefaction Curve
JCT 6-9-82
700
600
500
400
300
200
100
0
Refrigeration
Watts
0
20
40
60
80
100
120
140
Liquefaction, ℓ/hr

Figure 1

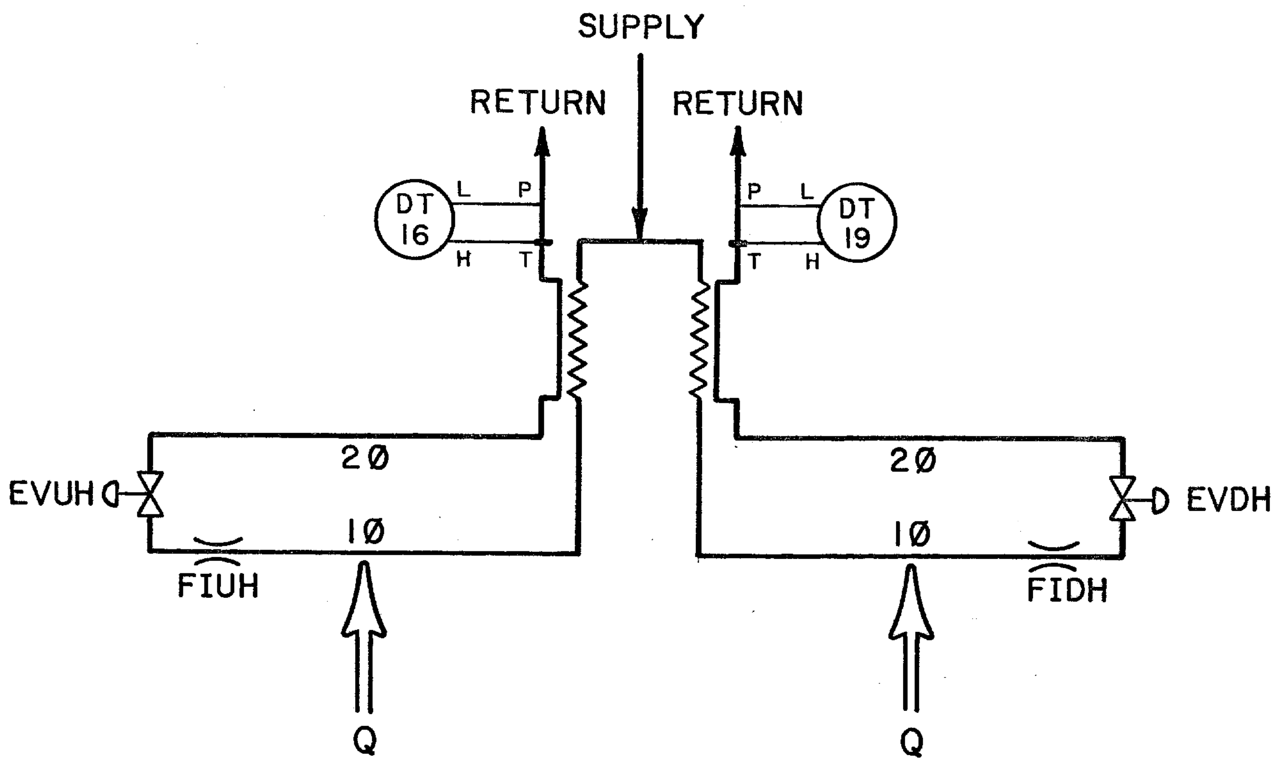

SUPPLY
RETURN
RETURN
DT 16
L
P
H
T
P
L
DT 19
T
H
2Ø
1Ø
2Ø
1Ø
EVUH
EVDH
FIUH
FIDH
Q
Q

Figure 2

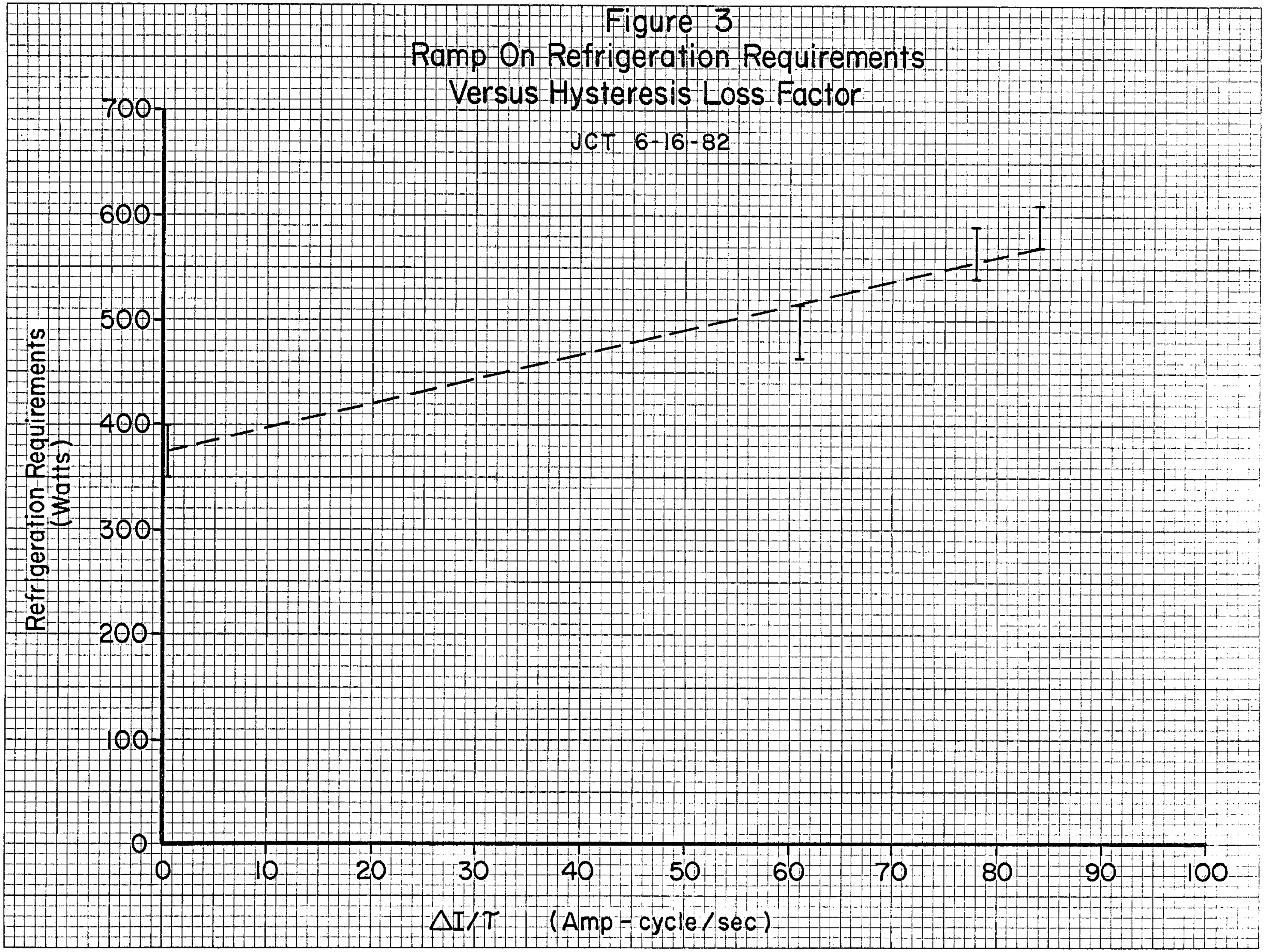


Figure 3
Ramp On Refrigeration Requirements
Versus Hysteresis Loss Factor
JCT 6-16-82
Refrigeration Requirements
(Watts)
700
600
500
400
300
200
100
0
0
10
20
30
40
50
60
70
80
90
100
ΔI/T (Amp-cycle/sec)

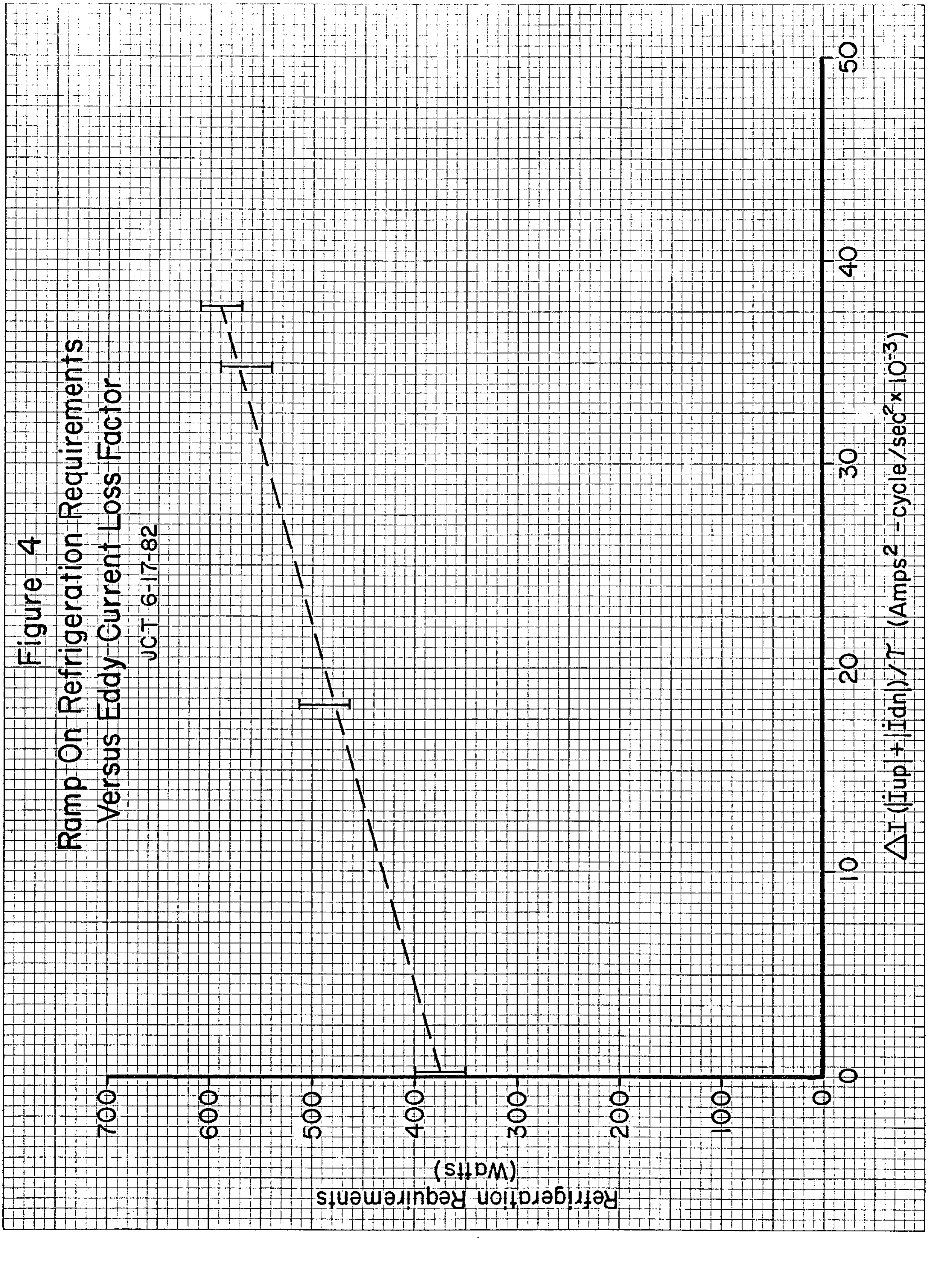


Figure 4
Ramp On Refrigeration Requirements
Versus Eddy Current Loss Factor
JCT 6-17-82
Refrigeration Requirements
(Watts)
700
600
500
400
300
200
100
0
0
10
20
30
40
50
ΔI (|İup| + |İdn|)/T (Amps² - cycle/sec² × 10⁻³)

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IX. HELIUM USAGE

C. H. Rode

The helium usage was much greater than expected, it averaged \$2000 per day. The usage is given by table 14-1 and Fig. 14-1. The figure shows that peak daily usage is 10 times "steady state leakage." The large peaks in Jan. and April were the start up of the transfer line. Feb. usage, which averaged \$3240, was driven by an endless list of repairs and an under estimation of the importance of transient leaks.

During the run, considerable magnet quenching was being done. We estimate that "A" Sector quenches contributed \$45K or 14% (note 4000 Amp quench: sector ∿\$3000, house ∿\$1000, cell ∿\$300). The largest single component was leakage. During April and May, a major effort was made to find and repair leaks; this resulted in decreasing the usage despite increasing quenching usage.

TABLE 14-1 Helium Costs per Day

	Month	\$/day	Liq. Liter/Hr.	SCFH
	Dec	\$ 851.	24.3	645.
	Jan	\$ 1559.	45.7	1211.
	Feb	\$ 3240.	92.6	2455.
	March	\$ 1784.	51.1	1353.
	April	\$ 2354.	67.3	1783.
	May	\$ 1687	48.2	1278.
	June (1-10)	\$ 1020	29.2	773.
Avg.	Jan-June	\$ 2063	59.0	1563

Note Cost: 5.5¢/SCF or \$ 1.4575/Liq. Liter (FY82 PRICE)

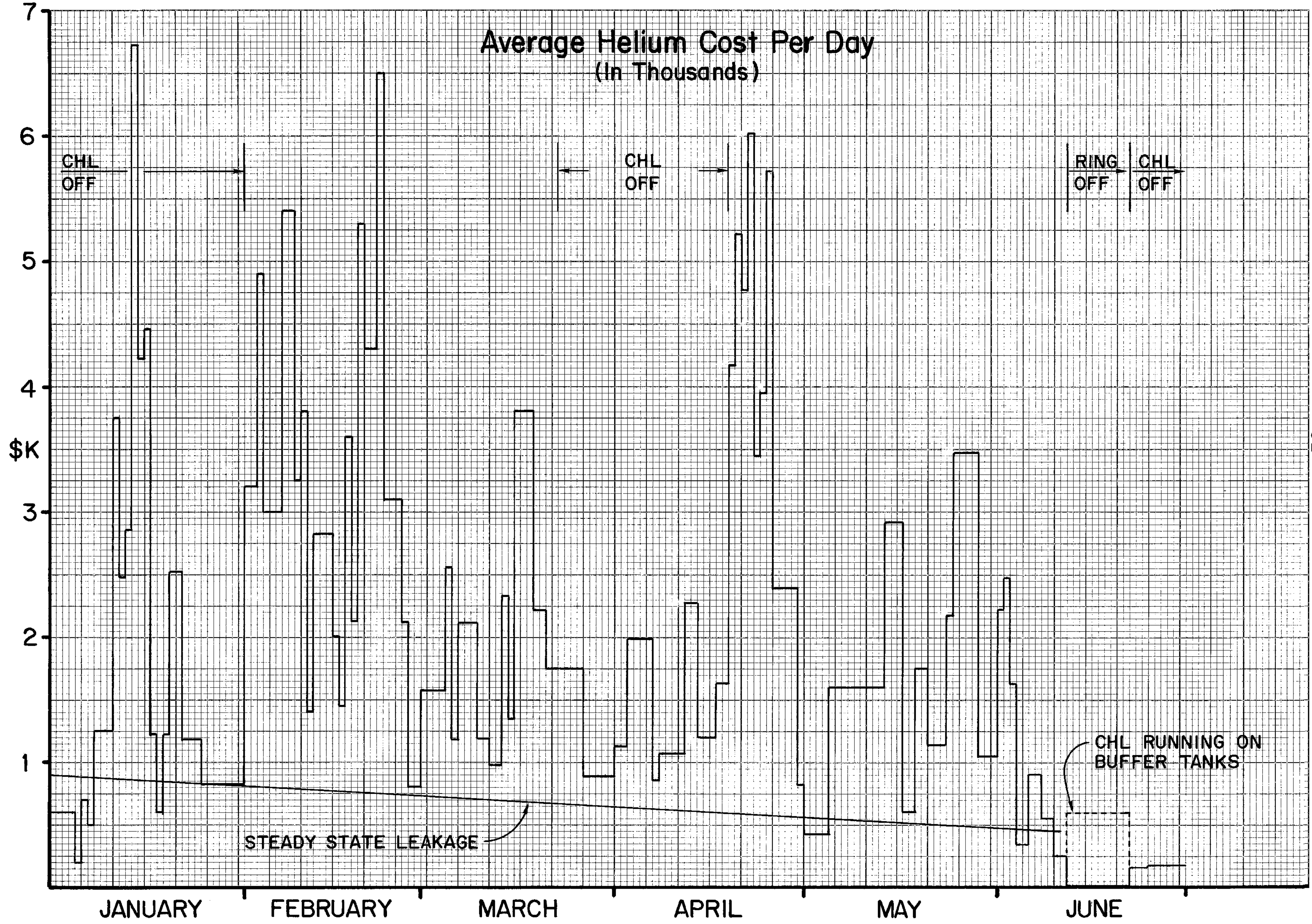
Average Helium Cost Per Day
(In Thousands)
$K
7
6
5
4
3
2
1
CHL
OFF
CHL
OFF
RING
OFF
CHL
OFF
CHL RUNNING ON
BUFFER TANKS
STEADY STATE LEAKAGE
JANUARY
FEBRUARY
MARCH
APRIL
MAY
JUNE

TABLE 14-II Breakdown of Helium Usages Jan-June 10, 1982
(Opinion of C.H. Rode and J. Theilacker)

	$K	%
Quenches		
Ring	45.0	13.6
B12	5.0	1.5
Steady State Leakage (not found or not repairable)		
Ring	59.1	17.9
CHL	49.2	14.9
Cryogenic Leaks		
Dry Expander Shaft Seal	4.0	1.2
Wet Expander Ring Failure	.5	.2
Cryogenic Valve Bellows	.8	.2
Bayonet Ball Valve	.5	.2
Power Leads	2.0	.6
CHL Rupture Disk	2.0	.6
Transfer Line Oscillations		
Jan	5.0	1.5
April	15.0	4.5
Malfunction of Reliefs (Inc. non reseating)		
8 in. Suction	10.0	3.0
1 in. Discharge	10.0	3.0
High Pressure Seal Leaks		
Ring	2.0	.6
Decontamination		
Compressor	4.0	1.2
Refrigerator	7.5	2.3
Heat Exchanger Derime	2.0	.6
Expander Warmup/Cooldown	7.5	2.3
Transfer Line	2.0	.6
Control Malfunction		
BØ Inventory Control Valves	5.0	1.5
μP	5.0	1.5
Compressor System Tripping Off		
BØ	10.0	3.0
CHL	5.0	1.5
Unaccounted For Usage	71.0	21.8
Total	330.0	100.0

At the end of the run, several days were devoted to loss measurements. Walker was able to get full system leakage of 800. +100. SCFH ($1056./day). After the 4200 Amp quench (Thur. 6/10/82 14:00), we again checked the suction reliefs to be sure they had reseated; two had not. "A1" 8 in. was leaking about 15 SCFH and "BT" 4 in. was leaking about 60 SCFH (total $99./day)

By Friday (6/11/82) 08:00, we had broken the system into four parts: A) CHL, B) 3 in. cross connect, C) transfer line plus E and F sector, 3) A-Sector including B12. The leakage of each part was independently measured.

A) CHL

Walker operated CHL making liquid into the heater first with one compressor then the other. The average leakage was $205./day.

B) 3 in. Crossconnect

The crossconnect was valved off at both ends and the pressure decay measured. The leakage was much less than $1./day.

C) Transfer Line Plus E and F Sector

The transfer line was warmed up to 50^{O}K, valved off at CHL with the return line being valved off at AØ. We then watched the pressure increase, then decrease, as the line warmed up. The leakage was $97./day. The majority of these leaks were in the temporary connections at E1, E2, and AØ.

D) A-Sector Including B12

The refrigerators were retuned to produce 20^{O}K refrigeration, which is a more stable mode, since it reduces the inventory in the system by a factor of 50. The leakage was $212./day of which a third of the leaks were repairable, a third were found but were not repairable, and the last third were not found.

Summary

At the end of the run, the sum of the steady state leakage was $451./day (22% of the total). This number projects to $1200./day for the full ring for FY82 He price. This number must be considered the lower limit for He usage, and will only be achieved if finding and fixing leaks is given the same poiority as delivering beam to the experiments. If the priority is that of preventive maintance in the conventional accelerator the He usage will be five times that. Assuming a compromise priority we get $2400/day (FY82 price for 1647ℓ/day) which is already 105% of the entire lab's He allocation.

Current Program

As a result of the A-Sector test we started many design improvement some of which we implemented during the run in order to keep operating. Using the same sequence as Table 14-II:

Cryogenic Leaks	
Dry expander shaft seal	: Peterson is redesigning Garner shaft seal.
Wet expander ring failure	: All expanders will be overhauled before start of run.
Cryogenic valve bellow	: A temporary seal to keep water out of bellows area is in use. Rode/Stoy are working on a design to add a piston ring to stop thermo accustic oscilations.
Bayonet ball valve	: --
Power leads	: A new design is installed
CHL rupture disk	: OK
Transfer line oscillation	
Jan. & April	: The back pressure regualtor was installed at E1; on the next run we will have the full ring therefore the back pressure regulator will be at CHL.
Malfunction of reliefs	
8 in suction	: --
1 in discharge	: we have piped it back into suction
High pressure seal leaks	
Ring	: we have ordered higher force accuators which will permit tightening the shaft seal on EVX1, EVX2, EVXP, & EVKI.
Decontamination	
Except for expander warmup/cooldown	: We have both the CHL purifier and at least one moble purifier that we will use during the run.

In order to deal with the transient leaks we will have two Tech I's dedicated to only leak hunting. For minor warm leaks they will repair them as they find them; all others they will simply report to the maintaince crew.

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X. POWER LEADS

J. Makara

Power to the superconducting magnets is supplied through American Magnetics[1] vapor cooled leads at the feed cans and special spool pieces. This section of the report discuss our experience with the leads.

These counterflow leads are cooled by venting liquid helium through the leads. Manufacturer estimates typical losses of 2.7E-3 liters per ampere hour per pair at optimum current, verified by our experience. Inadequate flow of helium cooling can result in electrical burnout and/or excessive heat load to magnet cryogenic system. Figure 1 shows a feed can power lead configuration; the spool piece design is somewhat different though functionally the same. The liquid helium flow of the magnet system is shown in a thick arrow, while the vent flow in a thin arrow.

Tests performed over the past several months concerning heat load to the magnet string are summarized in Table 1. Data shown gives the heat load as measured between the TI and TR in Figure 1 per power lead at a given vent temperature, with the required vent flow for such temperature. The "Buss Bar Only" configuration involves no additional cooling (or heating during non power conditions) of the lead flag by water, only tue venting helium is used as such. A top plate water heating system was added to reduce frost buildup around the flag and warm end seal area. Operation at -40°C vent temperature still produced failures of a G10 to G10 epoxy joint at the seal area. Warmer vent temperatures, say 0°C, reduces the icing and seal problems but the heat load to the string is too severe during non powered conditions. One test showed no subcooling in the magnet string until the power was turned on. The power-on condition required increased helium vent flow to remove the resistive heat generated in the leads, through still at same vent temperatures. During non power conditions, the vent flow is so reduced that the conduction heat load through the leads is to severe. A solution would require some form of external heat load at the top of flags to simulate power-on resistance heat load thereby forcing increased vent flow while keeping vent temperature constant, preferably above 0°C to prevent frosting.

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[1]American Magnetics Inc. Oak Ridge, Texas

The present workable option appears to be flag water heating and a new warm end seal design consisting of a weld joint, improved thermal insulation, and a threaded-epoxied G10 to S.S. seal. This configuration allows warmer vent temperatures limited by the voltage drop across the leads that can be tolerated. Tests showed that temperatures up to 18°C can be allowed without a high heat load. Flag water heating during non-power condition acts as a heat source forcing a change in the lead temperature profile and increased vent flow for same vent temperature. A heat load as low as 4 watts per lead at 6 liters per hour liquid helium equivalent was measured for non-power operation.

Present plans for vent temperature regulation involve using commercially available thermostatic expansion valves commonly used in refrigeration systems. Several of the tests were made using -40°C regulators, through most were performed with manual intervention. Future tests planned will attempt control by 0°C regulators.

LOAD FOR -40°C OUTLET GAS

CONNECTION TYPE	DC AMP 0	DC AMP 2000	DC AMP 3000	DC AMP 3500	DC AMP 4000	AC AMP 3000 (1658 RMS)	AC AMP 3500 (1935 RMS)	AC AMP 4000 (2211 RMS)
BUSS BAR ONLY	15BC / 15-25							
TOP PLATE WATER HEATING	24A 18B / 35	6 / B 25 HEATING	0 / B 50 HEATING	4 / B 50 HEATING		8±3 B 12 (2200A) E / B E 50 50	5±3 / B 40	
TOP PLATE & FLAG WATER HEATING	8 / C 55					±3	±3	
FLAG WATER HEATING ONLY								

A - A2 FEB, MARCH 82
B - B12 FEED 3-31 TO 4-2-82
C - B12 SPOOL 3-31 TO 4-2-82
* D - LAB 2 4-14-82
E - A2 APRIL 82, A2 MAY 82
F - B12 FEED MAY 82
G -
H -

DC WATT ± AC WATT / FLOW Scfh A
PER LEAD

LOAD FOR 0°C OUTLET GAS

CONNECTION TYPE	DC AMP 0	DC AMP 2000	DC AMP 3000	DC AMP 3500	DC AMP 4000	AC AMP 3000 (1658 RMS)	AC AMP 3500 (1935 RMS)	AC AMP 4000 (2211 RMS)
BUSS BAR ONLY	A							
TOP PLATE WATER HEATING	29A 22E 30E / 25A 20E 20E	B	<6 / B 50 COOLING	(3750A) 13 (COOL) / 80E	12 (COOL -20°C) / 85/90E	(2200A) 17 (COOL -15°C) / 35/40 E	9 ±3 / B 30	6±3 / E 50
TOP PLATE & FLAG WATER HEATING	12 / C 30	9C / C 50	9 4 / C 60	9 / C 60 HEATING		8 ±3 (COOL -10°C) / C 60	12 ±3 (HOT +10°C) / 40	
FLAG WATER HEATING ONLY	4±1D / F D 60	7 / F 50	D	7 (WARM 18°C) / D 50	F	D	3 / D 60	(5KA) 1.5 / F D 60

* McInturf, A. TM 1114

TABLE 1 (CONTINUED)

B12 (6-3-82)

VENT TEMP. (-6 TO +6°C) FLAG WATER HEATING

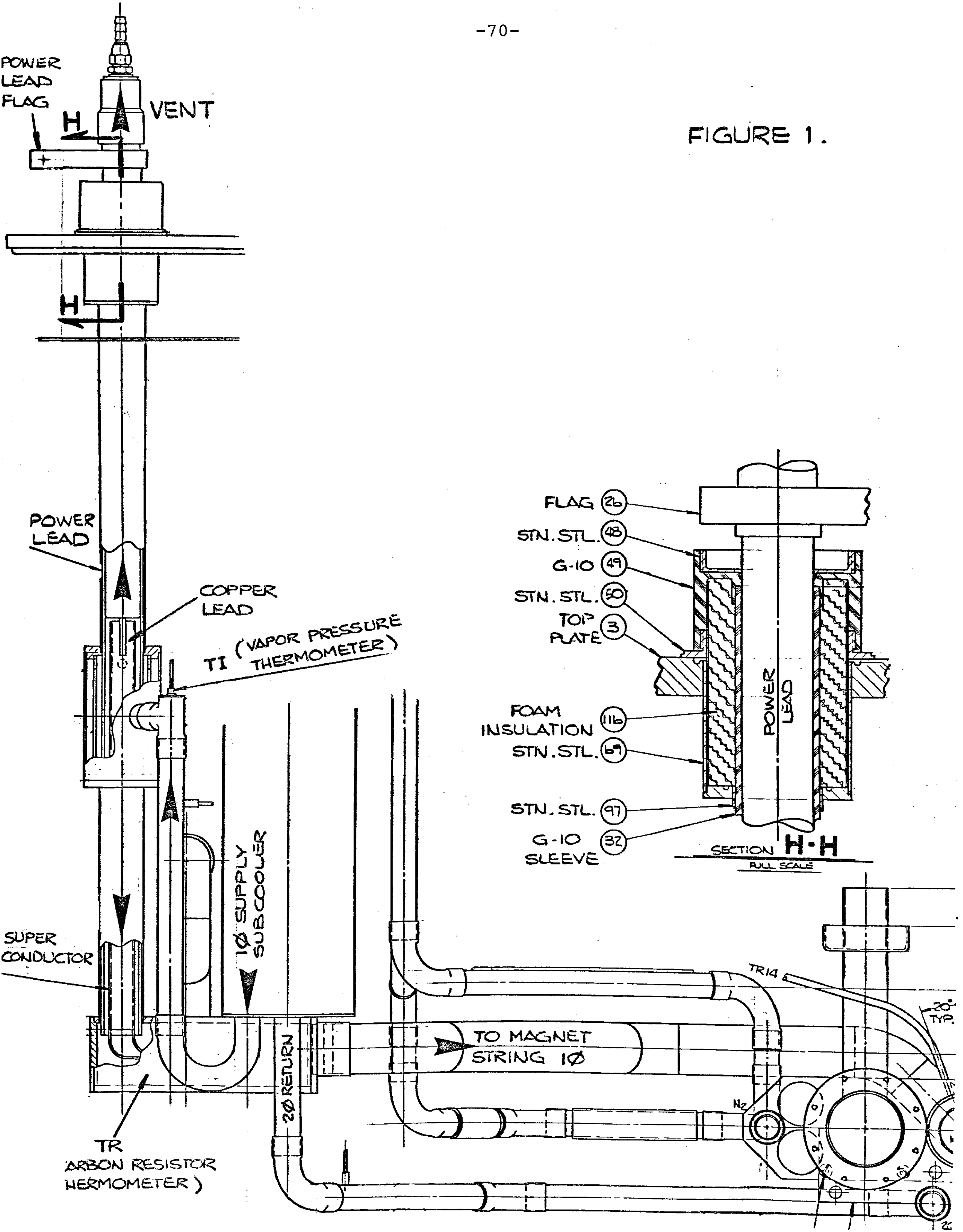
	1KA DC	1.5	2	2.5	2.5
U.S. (WITH LEAK)	4W / 40	6W / 40		12W / 40	8W / 45
D.S.	2W / 40	11W / 40	6W / 40	10W / 40	9W / 45

LAB 2 SPOOL (McINTURF, TM 1114)

18°C VENT

	0 DC	3500 DC	3500 AC	4000 AC
FLAG WATER HEATING ONLY	14W / 25	7W / 50	>8W / 40	3W / 50

FIGURE 1.


POWER
LEAD
FLAG
VENT
H
H
POWER
LEAD
COPPER
LEAD
TI (VAPOR PRESSURE
THERMOMETER)
1Ø SUPPLY
SUBCOOLER
SUPER
CONDUCTOR
TR
(CARBON RESISTOR
THERMOMETER)
2Ø RETURN
TO MAGNET
STRING 1Ø
TR14
20°
TYP.
N2
FLAG 26
STN. STL. 48
G-10 49
STN. STL. 50
TOP
PLATE 3
FOAM
INSULATION 116
STN. STL. 69
STN. STL. 97
G-10 32
SLEEVE
POWER
LEAD
SECTION H-H
FULL SCALE

XI. SUCTION HEADER TEST

J. Misek

The 8" suction header tests initially developed from the need to know the peak pressures in the 8" header during full sector quenches. The peak pressure values could establish the need to utilize the auxiliary tunnel relief valves. These valves had been included in the initial header design to limit the pressure rise to 45-60 psig., but it was hoped that their use could be avoided both from an operational/safety point of view, and the cost to fabricate and install them. Previous testing at B12 had shown that peak pressures were at levels which would require the auxiliary tunnel relief valves, though some doubt still existed because of the physical differences between the B12 and the tunnel header. The A2 cryoloop 8" header was instrumented with strain gage pressure transducers, temperature sensing diodes, and a linear motion potentiometer to measure the dynamic pipe movement due to gas flow and the movement due to temperature. Their locations in the A2 cryoloop are shown on Figure 1. Note that the pressure transducers at 21-1 1Ø and 28-5 1Ø were installed on the magnet side of the Kautzky relief valve to monitor peak pressures in the magnet cryostats. It is known from B12 testing that the actual cryostat peak pressure is some 15 psi above the pressure as measured with the pressure transducers ahead of the Kautzky relief valves. Signals from all the transducers were routed to an eight channel chart recorder, typical output is shown in Figure 2.

The pressure data collected from full sector quenches with magnet currents of 1500, 2250, and 2600 A led to the conclusion that the vertical leg from the tunnel to the A2 relief valve was the major item limiting flow out of, and increasing the pressures in, the tunnel header. The data also indicated that proceeding to 4000 A full sector quenches would, in all probability, cause header pressures to exceed 100 psig design pressure limit. The pressure data quench current dependence decided the issue in favor of installing the 45 psig relief valves. Testing continued with the relief valves installed, in full sector quenches of 3000 and 3350 A. Projections from the data for these quenches indicated that the 45 psi tunnel relief valves would maintain peak header pressures below 90 psig at 4000 A, see Figure 3.

The 8" header motion transducer full sector quench data was reviewed for motion along the 200 feet length between expansion joints in the direction of the gas flow. No motion was seen during the pressure spike in the header on full sector quenches and only 1/4" of movement was detected on an asymmetrical one house quench. That quench of cells between A15 and A25 tested the pressure drop across the loop in the 8" header at the feedbox region. A pressure drop of greater than 10 psi would have required additional lateral restraint at the feedbox regions, but no measureable pressure drop was detected.

Temperature changes of the header during full sector quenches were observed to be less than 50°K. This corresponds to the extension of each expansion joint by approximately 1.25" well within the 2.0" working range, if the three expansion joints in a 400 foot section equally share the contraction of the header. This sharing of contraction was verified during Kautzky valve "opening verification" tests when Kautzky relief valves were opened allowing cold gas to enter the header. Measurements of each expansion joint when the header was cold and when it had warmed showed strokes of 1/4" to 1/2" with each expansion joint returning to its initial position within $\pm$ 1/8".

The temperature data showed that the header temperature dropped below 220°K during the 3350 A full sector quench. The cause was a stuck Kautzky relief valve adjacent to the temperature sensor. This situation is similar to initiating a quick cooldown through the 1 spool relief valve. The header showed minimal movement in this occurrence.

Conclusions from these initial tests of the 8" header indicate that auxiliary tunnel relief valves will be required for full power operation of the Saver. These valves, set at 50 psig, will cause venting into the tunnel only for full sector quenches. The headers basic design features are adequate and anchors, braces, and brackets will be uniformly sized to allow 100 psig operation.

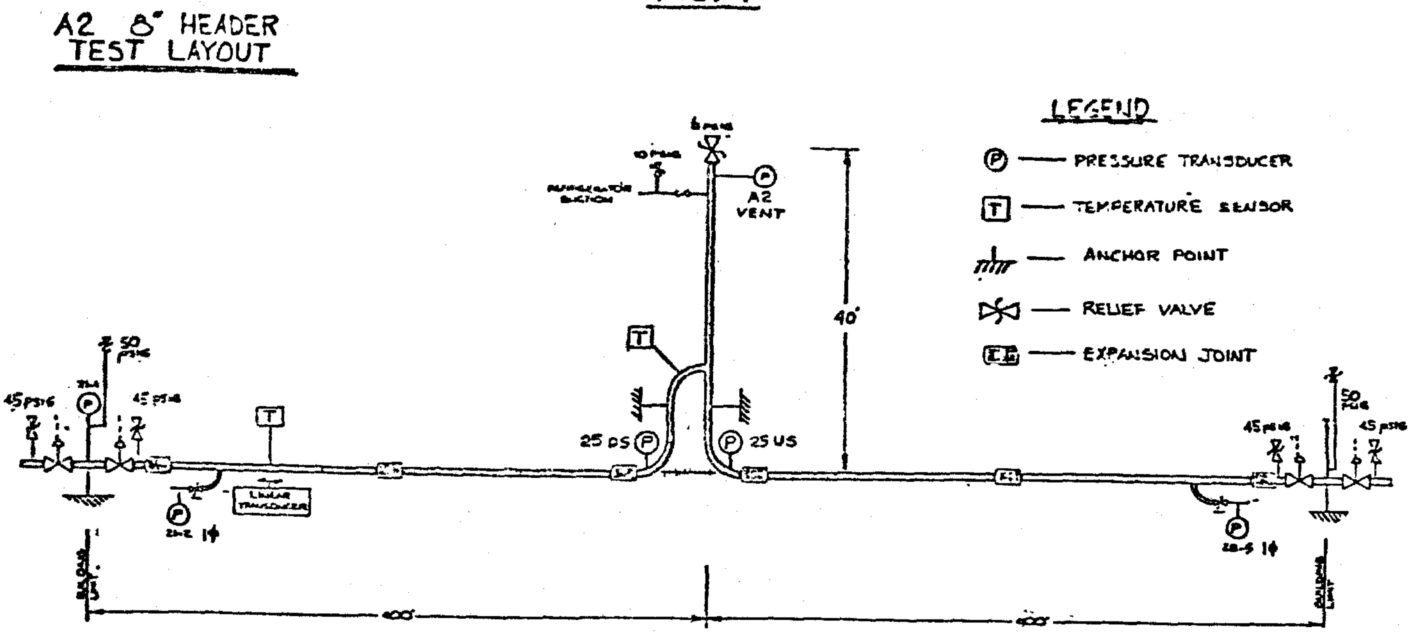
As a side note to these recent tests, preliminary cooldown tests on the 8" header were performed a year earlier. These earlier header tests monitored header movement as the liquid helium inventory of the A2 magnet string, approximately 700 liters, was dumped into the header through the cooldown valves located at each end of the magnet string. The time required to dump was approximately 35 minutes. During this time, each bellows was monitored visually for motion. The results verified that all bellows share pipe contraction in a given section and upon warming return to their initial state. Maximum bellows stroking was 1½". Another part of this test was to monitor header elevation to see if any vertical bowing of the header would occur due to differential contraction between the top and bottom of the pipe as first observed at B12. Bowing was observed at one end of the magnet string during the header cooldown. A 40-ft section of header lifted 2 in. immediately downstream of the cooldown valve. The opposite end of the string did not see this bowing. This is believed to be due to more turbulent flow and less stratification of the helium in the header. The region where the bowing was observed had no other flow than that from the cooldown valve. This was not true at the other cooldown valve region where both the refrigerator return flow and the warmed-up helium gas from the other cooldown valve passed through the header at this region on its way toward the compressor station. The magnitude of this bowing was not considered great enough to affect the header as bracket design can tolerate higher loading and flex hose vertical clearance is approximately 4 in. In addition, the probability of having the headers flow null point adjacent to the cooldown region should not be a usual occurrence.

In establishing an operating limit for the 8" header of 100 psig, consideration had to be given to the pressure drops in various locations under quench conditions as well as the peak pressures which generate gas flow. Aided by the A-Sector testing, the header was reviewed for operation. Table I lists principal elements in the tunnel 8" header installation and conclusions drawn from their analysis and review.

TABLE I

8" Header Element	Failure Mode	Failure Effect	Failure Pressure (Force)	Safety Factor	Maximum Working Pressure	Comments
I. Expansion Joint	Inelastic yield of convolutes	Loss of cycle life or leaks	200 psig	2 (with no squirm protection)	100 psig	Manufacturer states 120 psig max. working pressure
II. Flex Hose Assembly						
a. Braided Flex 2" & 3"	Rupture	Leaks	1000 psig & 1500 1500 psig	4	250 psig & 375 psig	
b. Spool Manifold	Rupture of alignment flex	Leaks	600 psig	4	150 psig	
c. Relief Valve Body	Fracture of flange lip	Leaks	12969 in-lb torque on flange	3 (with no rotation of valve body)	100 psig	Torque transmitted due to alignment flex.
III. 8" SCH 5 Pipe	Rupture	Leaks	1890 psig	5	378 psig	PER ANSI B 31.3
IV. 8" Tees & Elbows	Rupture	Leaks	---	---	158 psig	PER MSS SP43 fabricated welded tee
V. Feed Box Region						
a. Tie Bar	Yield of the bar	Excessive stroking of expansion joint and movement of pipe adjacent to feedcan	404 psig	4	101 psig	
b. Tie Bar Elbow Bracket	Deformation of elbow at bracket	Same as V.a.	468 psig	4	117 psig	
c. Lateral Support Structure	Slip of unistrut nuts	Same as V.a.	208 psi differential (6362 lbs)	4	52 psi differential	Assumes sharing of load on both sides of feed can thru tie bar
d. Tabs on 8" pipe vertical riser	Bending of tabs	Loading of adjacent expansion joints laterally until tie bar contacts feed can. Plastically deforms expansion joints. Possible leaks	306 psi differential	4	76 psi differential	
VI. Double Turn Around Region						
a. Anchor & Support Brackets	Slip of Header	Bending of cooldown piping. Possible leaks.	303 psi differential (9280 lbs)	3 (straps tightened to yield)	101 psi differential	Assumes sharing of load between 2 anchors & 2 supports
b. 6" Butterfly Valve	Rupture	Leaks	---	---	270 psi	Per manufacturer.
c. Fabricated Manifold 6" pipe	Rupture	Leaks	2234 psig	5	447 psig	Per ANSI B 31.3
VII. A12/F47 Ceiling Anchor	Anchor pull out	Excessive stroking of expansion joint possible leaks	496 psig	4	124 psig	
VIII. A12/F47 Wall Anchor	Same as VII	Same as VII	688 psig	4	172 psig	
IX. F47-3 Penetration Anchor	Pipe deformation	Excessive motion on expansion joint. Possible leaks.	468 psig	4	117 psig	
X. 11 & 49 End Anchor Assembly	Concrete anchor pull out	Excessive motion on expansion joint. Possible leaks.	660 psig	4	165 psig	
XI. Expansion Joint Guide Assembly	Bending of Guide	Excessive motion on expansion joint. Possible leaks.	1408 psig	4	352 psig	
XII. Pipe Support Assembly	Concrete anchor pullout	Excessive header movement & damage to flex hoses & any equipment underneath header.	4320 lbs. vertical 3840 lbs horizontal	4 4	(1080 lbs) vertical (960 lbs) horizontal	Supports are nominally every 10' or 100 lbs of header. 200' of header would have to lift to attain this loading vertically. Horizontal loading is nominally the rolling friction of rollers Less than 100 lbs.

FIG. 1


A2 8" HEADER
TEST LAYOUT
LEGEND
PRESSURE TRANSDUCER
TEMPERATURE SENSOR
ANCHOR POINT
RELIEF VALVE
EXPANSION JOINT
A2
VENT
40'
25 DS
25 US
45 PSIG
50 PSIG
1ϕ

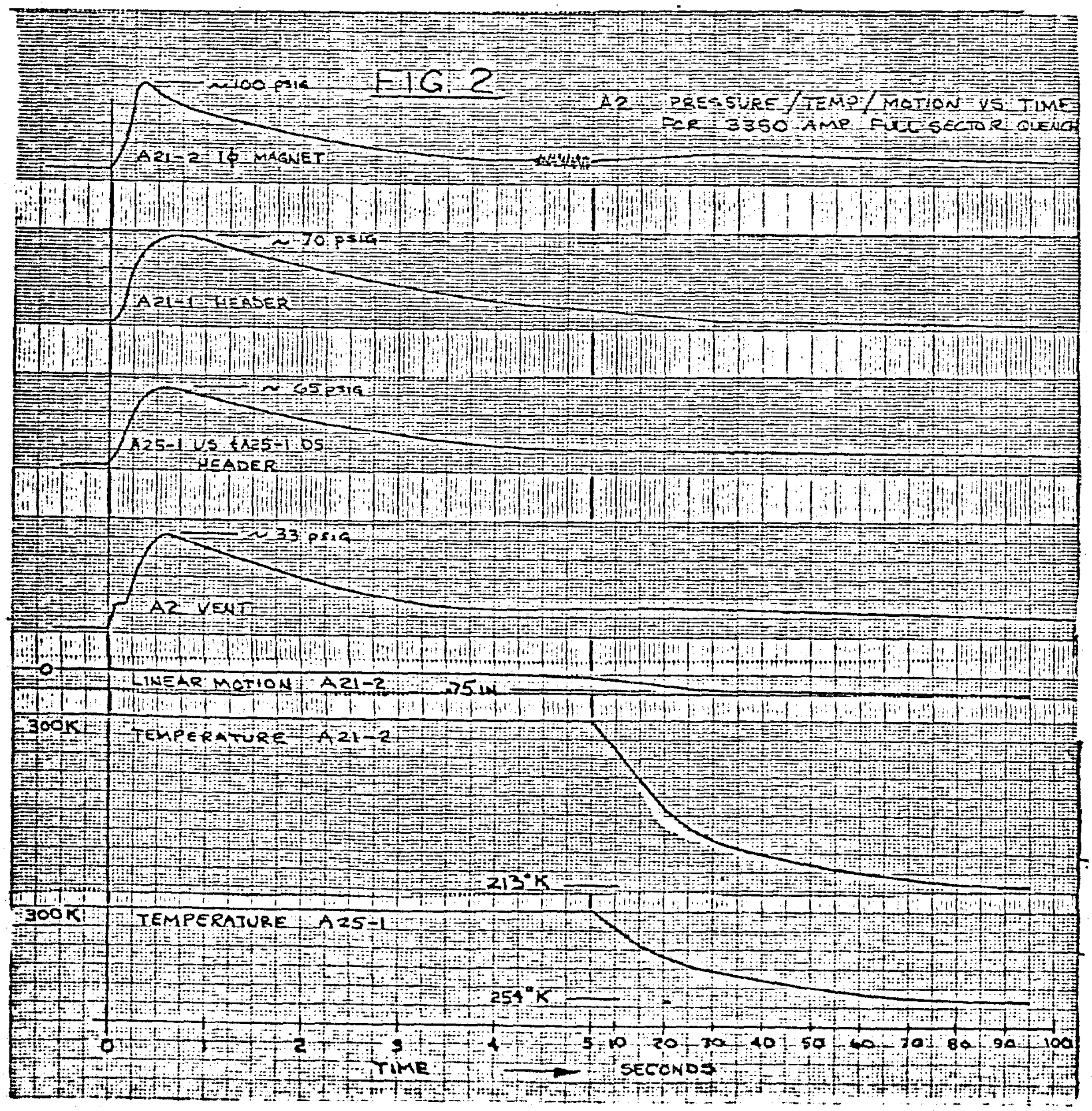
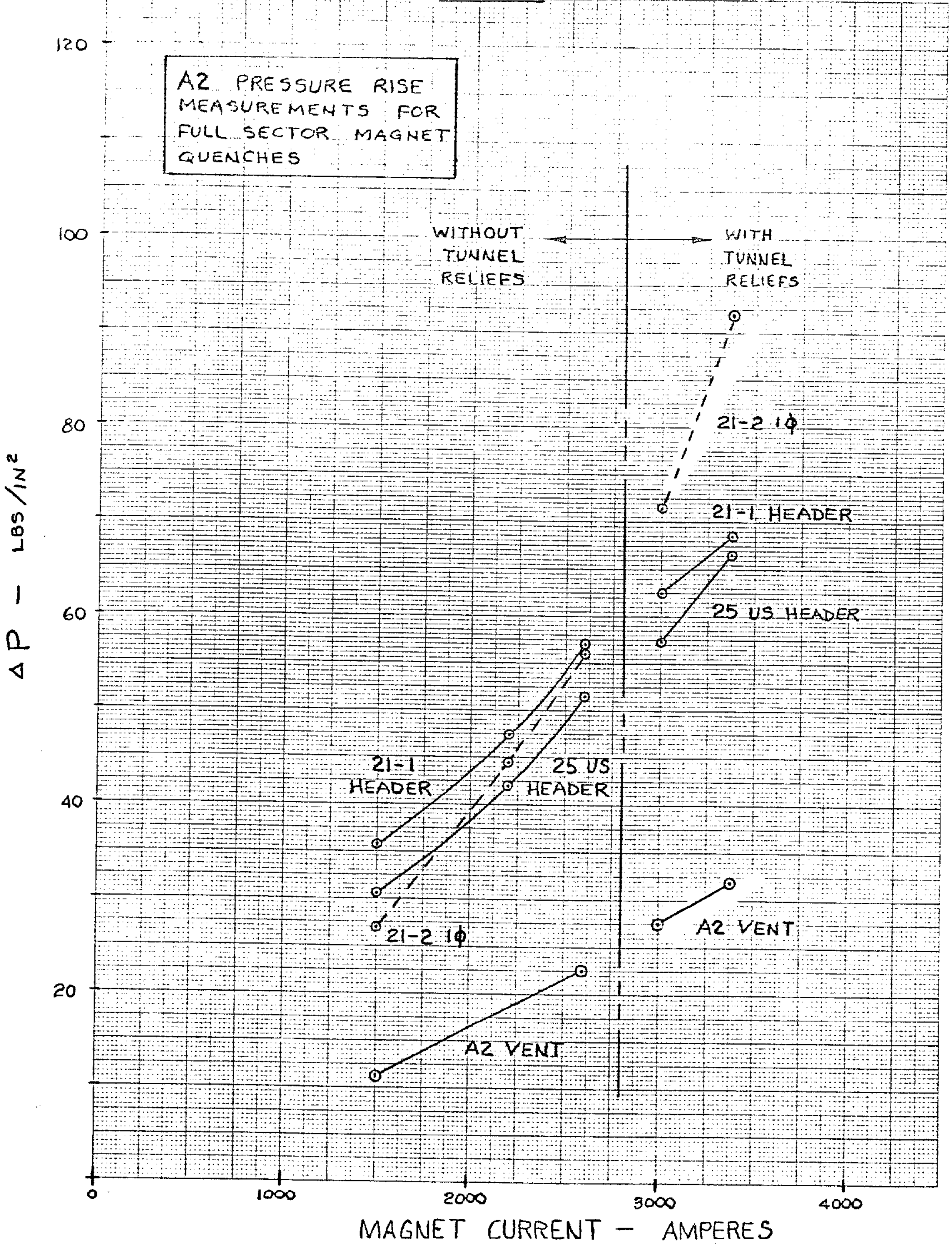


FIG. 2
A2 PRESSURE/TEMP/MOTION VS TIME
FOR 3350 AMP FULL SECTOR QUENCH
~100 PSIG
A21-2 1ϕ MAGNET
~70 PSIG
A21-1 HEADER
~65 PSIG
A25-1 US & A25-1 DS
HEADER
~33 PSIG
A2 VENT
0
LINEAR MOTION A21-2
.75 IN
300K
TEMPERATURE A21-2
213°K
300K
TEMPERATURE A25-1
254°K
0
1
2
3
4
5
10
20
30
40
50
60
70
80
90
100
TIME
SECONDS

FIG. 3


A2 PRESSURE RISE MEASUREMENTS FOR FULL SECTOR MAGNET QUENCHES
WITHOUT TUNNEL RELIEFS
WITH TUNNEL RELIEFS
21-2 1φ
21-1 HEADER
25 US HEADER
A2 VENT
21-1 HEADER
25 US HEADER
21-2 1φ
A2 VENT
120
100
80
60
40
20
0
1000
2000
3000
4000
ΔP — LBS/IN²
MAGNET CURRENT — AMPERES

XII. SHELL SIDE PRESSURE DROP

J.C. Theilacker

INTRODUCTION

A refrigerator study was performed to investigate the exchanger shell side pressure drop. The three refrigerators operating during the recent testing (A1, A2, A3) all showed different shell side pressure drop characteristics. The A1 heat exchanger (a prototype which will be replaced) displayed the highest pressure drop (approx. 8 psi) under normal operating conditions (NOC). This is due to improper sized manifolding in the #3 exchanger in this prototype heat exchanger. The A2 vertical heat exchanger produced the lowest pressure drop under NOC while the A3 horizontal exchanger fell in-between. This study was directed toward the A3 horizontal heat exchanger since it is incorporated in all but two of the satellite refrigerators. The objective was to determine why there was a difference between the horizontal and vertical exchanger pressure drops.

Minimizing the shell side pressure drop is important in order to achieve a maximum current carrying ability of the magnets. This is because the shell side pressure drop determines the magnet 2Ø pressure (∿ suction pressure + shell side ΔP) and as well as the 2Ø and 1Ø temperature. Figure 1 shows the 2Ø temperature of helium as a function of pressure. In the area of interest, a 1/2°K decrease in temperature will allow ∿10% increase in magnet current.

TEST RESULTS

A test was performed to measure the horizontal heat exchanger shell side pressure drop as a function of flow rate. The refrigerator operated in satellite mode with no current being applied to the magnets. During the test, the flow from CHL was adjusted in attempt to maintain a constant temperature profile in the heat exchanger. Results are shown in Figure 2.

As expected, the pressure drop was proportional to the square of the mass flow rate. At the design refrigerator capacity of 966 watts the pressure drop was 3.4 - 3.8 psid. Data was taken on 3-3-82 at intermediate points in the system in order to localize the pressure drop. For these conditions a 1.5 psid was measured across the return U-tube and 0.6 psid in the valve box.

Table 1 compares the pressure drop of a vertical and horizontal exchanger. Both refrigerators were in stand-alone mode operating at nearly the same condition. The data is presented only as a first order comparison, since it is difficult to estimate the mass flow rate through the dry engines. The flow rate used was 30-36% above design capacity in order to maximize the resolution on the pressure measurement. A 50% higher ΔP was found in the A3 refrigerator at these conditions. Typically, differences between the horizontal and vertical exchangers were not this large.

The last test was performed prior to the 4200 Amp. run. We needed to lower the magnet temperatures, especially at cell 19 where we were having 4000 Amp spontaneous quenches. We attempted to lower the 2Ø pressure (and thus temperature) by reducing the temperature throughout the exchanger. This would lower the heat exchanger pressure drop inversely proportional to the change in density. During this test the helium flow through the refrigerator was considerably above design flow.

To reduce the exchanger temperature profile we increased the heat exchanger imbalance by taking more flow from CHL. Liquid helium flows from CHL were 400, 200, and 200 ℓ/hr. for A1, A2, A3 respectively. This reduced the shell side ΔP to 2.0, 2.6 and 4.3 psid. for A1, A2 and A3 respectively.

The extremely large change in ΔP at A1 is because more than half of the normal ΔP is in the #3 exchanger, which is very sensitive to flow imbalance. We believe that the lack of change at A3 is due to large components of the ΔP being at $5^{O}K$ and/or $300^{O}K$ ends of the refrigerator, which are unaffected by flow imbalance.

Recommendations

One further test should be performed to determine how much of the horizontal heat exchanger ΔP is at the $300^{O}K$ and $5^{O}K$ portions of the plant. If one component dominates, it may warrant corrective action.

Keywords:

Exchanger

Heat exchanger

Pressure Drop

Shell Side Pressure Drop Comparison

A2 - A3

High Flow Conditions

4-30-82

	FI4	FIUH	FIDH	Dry	Total	TI7	PI11	PI16	PI19
A2	73+ g/s Pegged	35 g/s	31 g/s	CTI 8 g/s est.	74 g/s	18 °K	5.2 psig	5.8 psig	6.0 psig
A3	73+ g/s	28 g/s	31 g/s	G-F 19 g/s	78 g/s	20 °K	7.9 psig	7.8 psig	8.1 psig

Total shell side flow

Middle of plant temperature

Shell side pressure from tunnel

2Ø pressure in magnets

Table 1

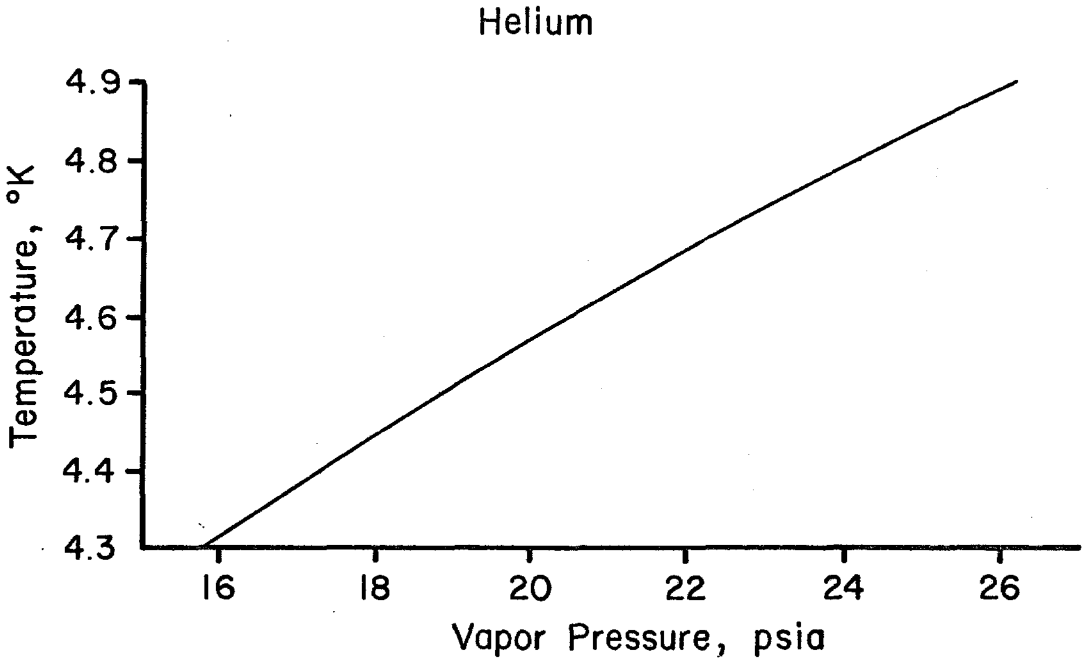
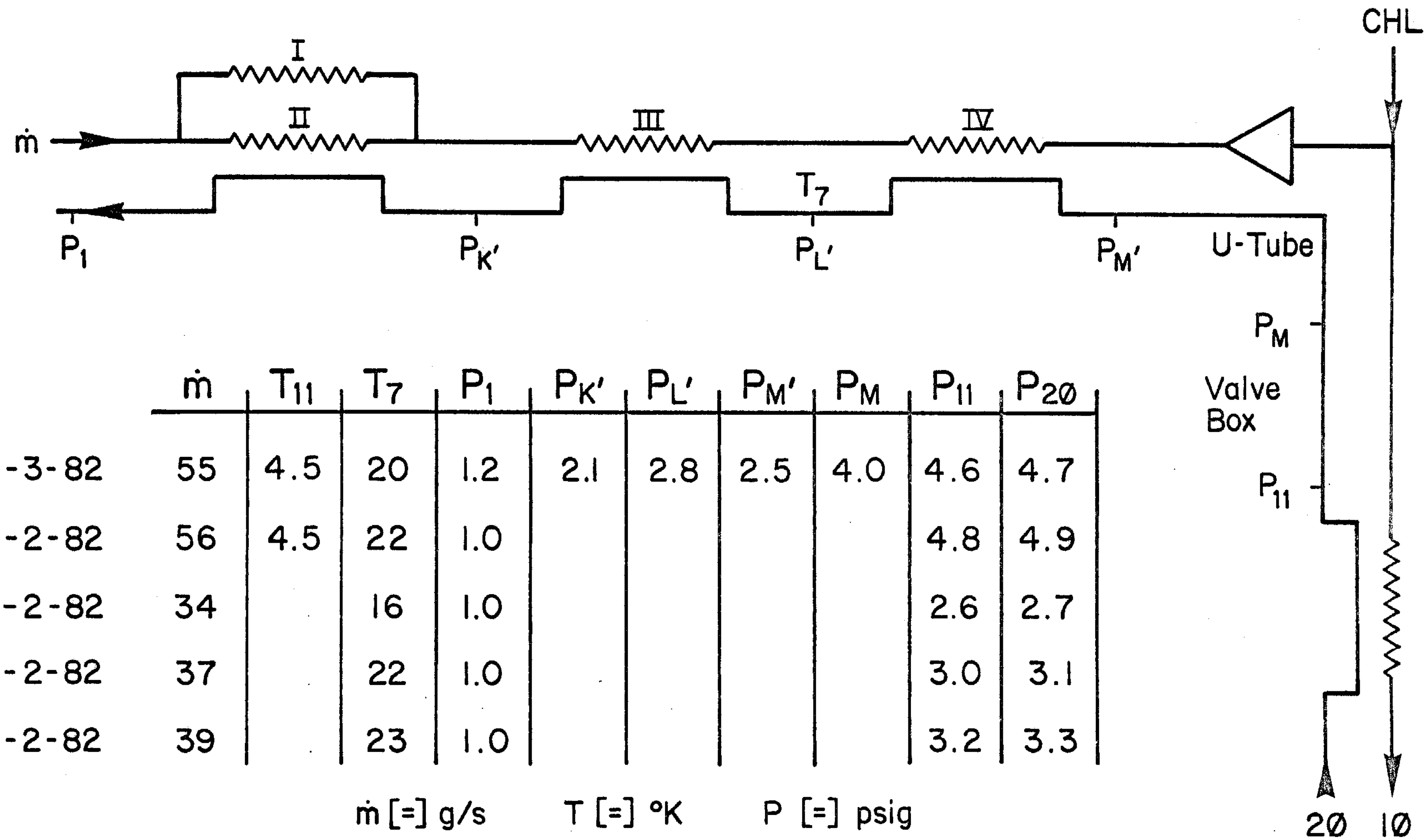

Helium
4.9
4.8
4.7
4.6
4.5
4.4
4.3
Temperature, °K
16
18
20
22
24
26
Vapor Pressure, psia

Figure 1



	$\dot{m}$	T_{11}	T_7	P_1	$P_{K'}$	$P_{L'}$	$P_{M'}$	P_M	P_{11}	P_{20}
3-3-82	55	4.5	20	1.2	2.1	2.8	2.5	4.0	4.6	4.7
6-2-82	56	4.5	22	1.0					4.8	4.9
6-2-82	34		16	1.0					2.6	2.7
6-2-82	37		22	1.0					3.0	3.1
6-2-82	39		23	1.0					3.2	3.3

$\dot{m}$ [=] g/s $\quad$ T [=] °K $\quad$ P [=] psig

A3 Shell Side Pressure Drop
Satellite Mode

Figure 2

Fermilab

XIII. COMPUTERS, ACTUATORS AND TRANSDUCERS

J. Gannon

Computers

The multibus-bus based computers performed very well during the run. Some initial difficulties with multiplexers on the A/D convertor cards were completely overcome after installation of an improved input protection network at each card.

Spontaneous reboots of computers was a difficulty initially in the early part of the run for unknown reasons but improved after installation of the multilayer M080 board.

There were no failures with any other cards in the six computers operating during this period. Communications with the computer were disrupted whenever the service building temperature was warmer than 90°F. or colder than 55°F. This had to be monitored closely during the run.

Actuators

The tunnel actuators used on the cooldown and JT valves operated as expected and without failures.

Several of the satellite and compressor actuators had difficulties. The cold box JT and bypass valves at A2 on the vertical heat exchanger had condensed water dripping on the motors. This caused a paste-like substance to form around the brushed and causes them to stick as they wore. The motors in these locations will be shielded in the future to alleviate this problem.

The only other actuator problems were with the high pressure warm valves in the compressor and refrigerator buildings. These valves require a larger than designed operating force. A stronger actuator will be installed in these locations in the future.

The other major difficulty with actuators has been mounting and spanning the linear actuators on the valve and/or replacing a failed valve (there were many). A coupler has been designed and procured to allow easy installation, adjustment, and replacement of valves in the refrigerator and compressor buildings.

Transducers

The gauge pressure transducers for measuring single temperatures and system pressures survived the rigors of the sector test without any know failures due to overpressure, temperature cycling, or pressure cycling. A few transducers failed shortly after start-up due to infant mortality in the electronics package.

The differential pressure transducers which measure the 2Ø superheat sometimes leaked to the reference port and/or shifted the zero offset when exposed to high pressure cycling during charging of the VPT or operation.

Discussions with the manufacturer have resulted in a modification to remedy this problem for existing and future units.

XIV. KAUTZKY VALVES

C. T. Murphy

Introduction

The A-Sector Test afforded the opportunity to test a large number of the Mark III version of the Kautzky valve[3] under the duress of a small number of high-pressure quenches and true "field" conditions. This testing compliments the tests at MTF, where a few valves (about 15) have endured many hundreds of quenches, mostly in the shunted dump resistor mode, i.e., at moderately low pressures. The quantitative result of the failure statistics gathered is that a Mark III valve has a 1.2% probability of failing to close, after opening during a quench, in a manner that requires either immediate or next-down-day access to the tunnel. However, valve improvements already manufactured in the Mark IV version of the valve, magnet production improvements which reduce the number of screws, nuts, washers and G-10 pieces burped out through the valve by the magnets during a quench, and improved controls on the alignment and torque-down procedure on the manifold connecting the two spool piece valves to the 3" flex hose, should reduce this failure probability to about 0.2%. (See Fig. 1.)

The fact that flaws in both the Mark III valve and the early magnets installed in A-Sector required 13 valve changes finally forced us to invent, test, document, and train technicians to perform the sensitive procedure of changing a valve under "cold" (20°K) conditions. The procedure is quick, has been mastered by about six people, and declared adequately safe by Accelerator safety personnel. Failure on one front has led to success on a different front.

Valve Failure Statistics

The details of the failure statistics are as follows.[4] There were 146 valves in A-Sector mounted on the 1Ø or 2Ø relief ports which open during quenches above about 1000 amps in current. Each valve opened at least 9 times and at most 22 times, for a total of 1992 valve openings, during which 2Ø helium flowed violently through the valve. The total number of valve failures (including twice those valves which failed twice) was 24. These failures can be tabulated in two groups:

Failures which are already cured for the next run:

No. of Failures	Cause
7	Broken stem welds in valve
4	Leaks to tunnel through O-ring on spool piece 1Ø valve

Failures which will eventually be cured:

No. of Failures	Cause
1	Poppet improperly secured to foot
9	Valve cured without replacement by torching and "popping"
3	Valves removed after failure to reset after torching and popping, had magnet hardware imbedded in poppet

Discussion

These failures warrant some discussion. The first one, "broken stem welds", was anticipated and already cured at the design and fabrication level before the A-Sector tests, but we had to use existing, obsolete valve actuators. When a stem weld failed, the valve failed unsafe (closed, will not open during a quench). This failure was detected by a sudden increase in the flow from the control pressure bottles[1] (a leak of warm helium into the 1Ø system through the broken weld.). In the Mark IV version currently being assembled, weld failures can still occur, but the valve will fail safe (i.e., open).

The second failure mode, "leaks to the tunnel through O-ring on the spool piece 1Ø valve" were clearly a result of very hasty installation of valves last summer. The leaking joints were found to have Marmon clamps which were extremely loose, and in all four instances, the leaking joint was the 1Ø joint between the spool piece flange and the Kautzky valve. The two Kautzky valves on the spool piece (1Ø, 2Ø relief) connect to a common manifold and must be properly aligned before final torquing of the Marmon clamps. This alignment and torquing procedure will be more carefully controlled in E and F Sectors hereafter.

Nine valve failures to reseat with leaks through the poppet were solved by a combination of thawing the frost with propane torches and "popping" the valves by removing momentarily the control pressure line. "Popping" the valve open was intended to flush out removable debris trapped between the poppet and the aluminum body, if trapped debris was the cause of the valve not fully closing. Another possible cause of a valve failing to close fully when cold is called the "alignment problem". If the actuator shaft is not concentric with the body seat by an amount between about 0.020" and 0.030", the valve will seal well at room temperature (because of the loose fit between the polyethylene poppet and the steel foot), but will not seat well cold because the loose fit has become a tight fit. By torching the aluminum body until it is a little above room temperature, and hopefully warming the poppet, then "popping" the valve open once gives it a chance to reseat warm.

Of the nine failures, at least four were definitely caused by trapped debris. This is known because later inspection of the poppets showed clear "footprints" of indentifiable objects on the poppet: 6-32 threads or a star washer. At least one was clearly a misalignment problem: the valve reseated suddenly while we were torching it, with an audible "crack". The other four instances can be attributed to either problem. Two of the poppets had definite notches on the seating surface (but so do most valves which come back from MTF after hundreds of cycles, none of which failed to reseat after a quench). The other two poppets appeared quite unscarred.

Three valves failed to close after torching and popping and had to be replaced immediately. All three had loose magnet hardware firmly wedged between the poppet and the body: a piece of G-10, a 6-32 screw, and a 6-32 nut. This hardware was used prior to magnet number 528 in the instrumentation lead tie-down system. Since then this hardware has been either riveted, "staked", or secured with Locktite 404 (see memo from J. Carson to T. Murphy, dated April 14, 1982). However, most of the magnets in A-Sector have serial numbers below 528.

By way of contrast, 16 valves of this same version have endured a total 8800 quenches at MTF without a single failure of this kind - failure to reseat after a quench. At MTF, the magnets all had serial numbers greater than 528. Both the MTF experience and our own data (indicating that as many as eight of the nine failures <u>might</u> be the result of trapped debris) suggest that the failure-to-close problem will eventually go away in the tunnel.

The most troubling failure occurred only once. A poppet became detached from the actuator foot, failing unsafe (i.e., the valve remained closed during several quenches). Post-mortem analysis by Tom Peterson revealed that the snap-ring had not been properly set in the groove of the poppet, came loose, and after being once squashed into the poppet (its footprint was clearly identifiable), was flushed into the header. The groove in the poppet showed no signs of having broken away, which led to the conclusion that the snap-ring was never set fully in the groove. This failure is troubling because it was discovered only by chance. It was discovered while searching for a broken stem weld, and was discovered because the valve failed to open when the control pressure was removed. We know that it had been failing to open for several weeks because we had noticed that the actuator of this valve was badly "domed", a result of the extra high pressures from the unrelieved magent. This failure points out the need to install Klixons on each valve - as planned - to prove in each quench that every Kautzky valve opened.

Conclusions

In Fig. 2, the valve failure rate is plotted as a function of cumulative valve openings. The graph suggests that the rate of failure was declining towards the end of the run. If this decline is real, it would confirm Koepke's qualitative observation at B-12 that most of the loose debris in magnets is blown out during the first ten or so quenches.

By way of conclusion, let us make a somewhat speculative projection of the failure rate expected in the next cryogenic run, the E-F Sector Test. Weld stem failures will not occur, nor will leaks through O-rings to the tunnel. Let us assume that failures to reseat because of trapped debris will be reduced by a factor of three as a result of improved magnets, and that the valve alignment problem is eliminated (as we currently believe). The resulting failure rate would then be about 4/1992 = 0.2% per valve opening, or 2.8% per full-cell quench. If there are ten full-cell quenches per day, one would then expect to have to service a Kautzky valve about every four days.

Procedures For Changing Valves

Procedures and special tools were invented (see ref. 2) during the A-Sector run which permit changing Kautzky valves when the 1Ø system is at 20°K or warmer and 2 1/2 psig or lower. The main hazards of the procedure are the momentary cloud of cold gas emitted from the 1Ø relief port while it is open and the possibility of emptying the warm header helium into the tunnel if it is not promptly capped. The job has been classified as an ODH class 2 job. At the moment, the job requires two trained technicians who do all the work and two observers who are needed for communication with the Tevatron console by walkie-talkie relay. A valve change takes a total access time of about 1 1/2 hours, but only 20 minutes for the actual changing procedure.

References

1. C. T. Murphy, "Doubler Kautzky Valve Supply Manifold Adjustment and Bottle Changing", Operations Bulletin No. 877, April 9, 1982.
2. C. T. Murphy, "Rules and Procedure for Kautzky Valve Change in Tunnel under Cryogenic Conditions", to be published.
3. "The Kautzky Valve", Fermilab Reports, June, 1982.
4. C. T. Murphy, "Kautzky Valve Problems in Tunnel", bulletins no. 1 through 6.

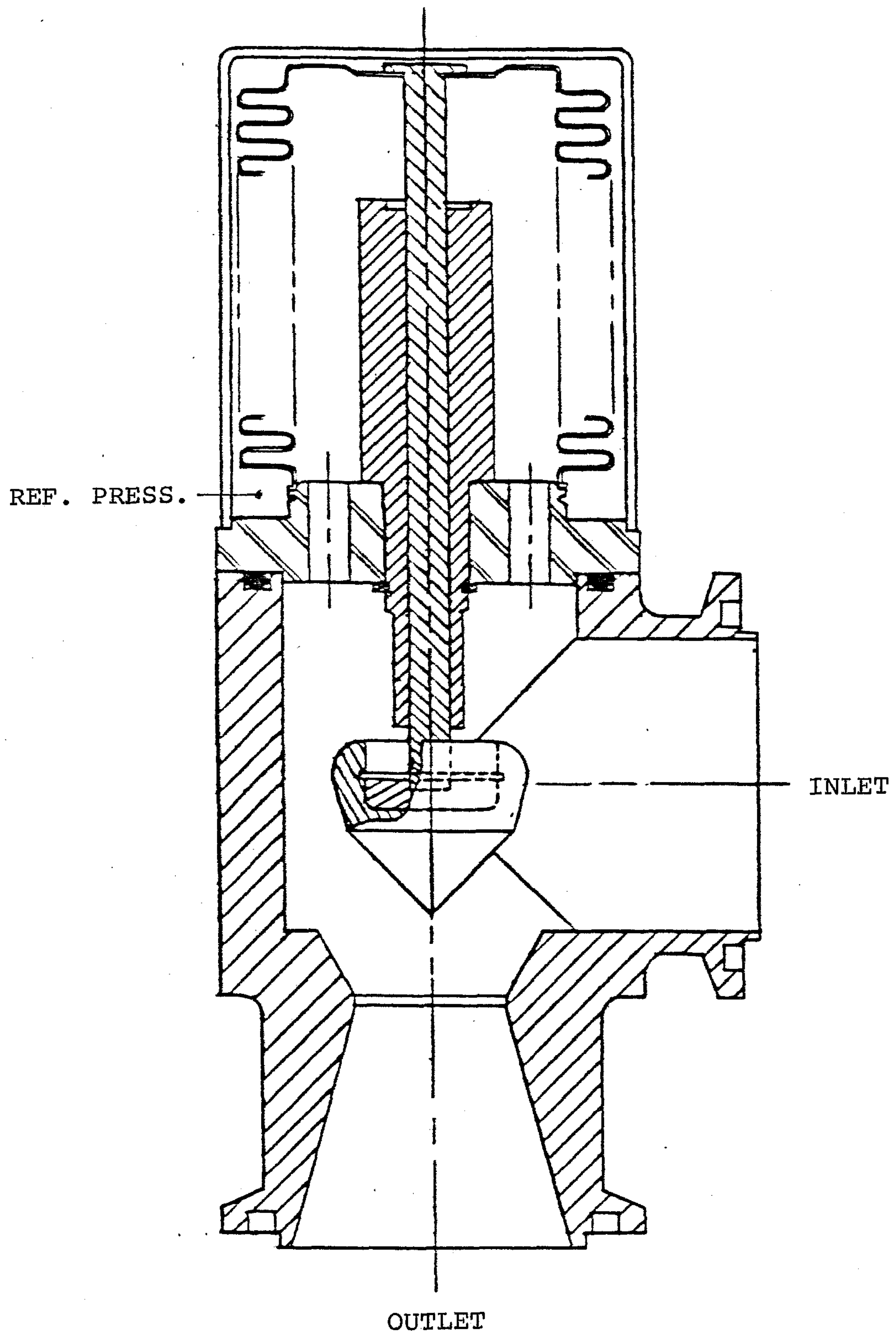
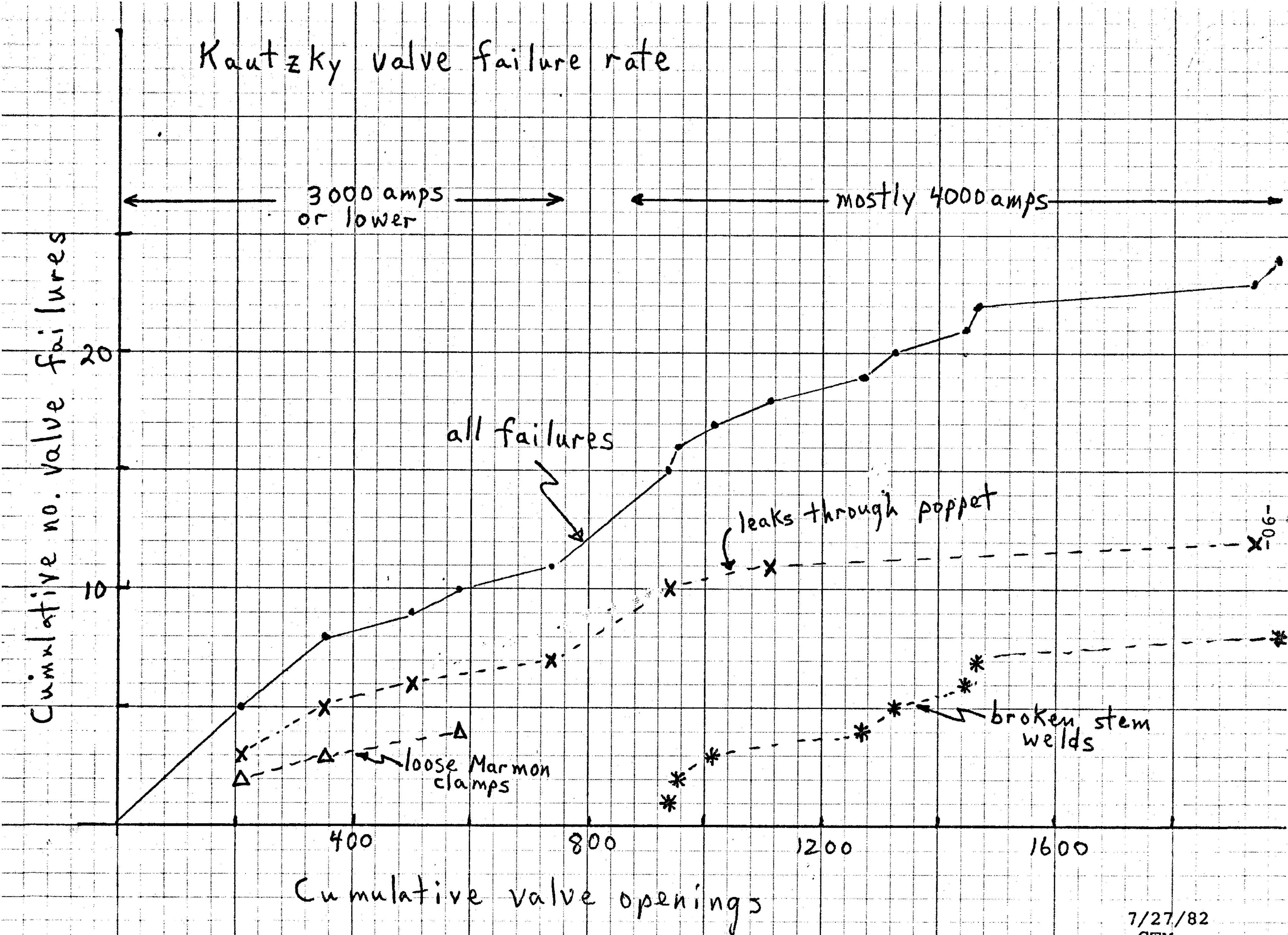


Figure 1. Kautzky valve.



Kautzky valve failure rate
3000 amps or lower
mostly 4000 amps
all failures
leaks through poppet
-90-
loose Marmon clamps
broken stem welds
Cumulative no. valve failures
10
20
Cumulative valve openings
400
800
1200
1600
7/27/82
CTM

Figure 2.

XV. AUTOMATIC QUENCH RECOVERY

Manuel I. Martin

0. GENERAL

The Automatic Quench Recovery (AQR) software was extensively tested in the A-sector. Seventeen (17) controlled quenches and one spontaneous quench were monitored and recovered using the AQR.

After a quench took place the system was considered fully recovered when

a) both magnet strings were full of liquid helium,

b) there was subcooling (≥6 psig) in the single-phase input of both strings, and

c) the measurement of superheat (DT's) was stable and below the set point.

The AQR consists of three distinct software-defined protocols:

a) Fast Quench Response (FQR) is an interrupt driven software package. Once a quench has been detected the FQR opens the proper Kautzky valves, inactivates the proper JT and CD cooldown loops, closes the JT valve(s) and, if requested by the protocol, opens the corresponding CD valve(s). Finally, the FQR activates the proper Quench Recovery Finite State Machine(s) (QRFSM).

b) Quench Recovery (QR) is a Finite State Machine whose function is to orderly close the Kautzky valves when the cool wave reaches predefined points within the magnet's string. When all the Kautzky valves are closed, the QRFSM activates the proper Cool Down Finite State Machine (CDFSM) forcing it to the Transition Mode.

c) Once the CDFSM is activated the system will go through Transition Mode to Fill Mode and then to Operate Mode. The CDFSM will control the JT and CD valves directly until entering the Operate Mode, at which time all loops are reactivated.

1. TESTS PERFORMED

The following table presents a summary of the tests. The table is organized by house and by chronological order.

TABLE A

House	Cell	Recovery Time (Minutes)	Comments
A1	A13	120	*Relief valve at A3 popped
	A19	48	
	A15	30	
	A13 & A17	50	
	A17	40	False quench
	A19	60	4200A
A2	A22	45	
	A24	16	
	A26	84	*μp problems
	A24	25	
	A22	50	
	A24	69	*μp rebooted
	A26	33	
	A28	56	
	A24	22	
A3	A34	115	*Fill mode premature
	A32	29	
	A36 & A34	60	

Note: For more detail, refer to the report by P. Martin and G. Tool, TM-1134.

From the table we obtain the following statistics for recovery time:

a) With all values included

A1	median	58 min.	standard deviation	32
A2	"	44.4 min.	" "	23
A3	"	68 min.	" "	44

All houses median = 52.9 min. with standard deviation of 29.3

b) Deleting the values marked with (*)

A1	median	45.6 min.	standard deviation	11
A2	"	33.3 min.	" "	15
A3	"	44.5 min.	" "	22

All houses median = 44.8 min., standard deviation 19

Figures 1 and 2 show two recoveries from induced quench at Cell 1 of A1. It must be noted that these figures correspond only to the first part of the recovery when the string enters the Operate Mode but the DT's are not stable yet.

2. OBSERVATIONS

During the 18 quenches where the AQR was used the following observations were made:

2.1 The FQR performed as designed with one exception where the proper tunnel JT valve did not close completely. The apparent cause was a frozen valve.

2.2 The FQR should have control over KV1 in order to speed up recovery. (Note: At A2, 3, 4 it is located on the upstream dipole.) (See Figs. 3,4 for location of valves.)

2.3 Although FQR has control over KV9 (KV10 at A1) this valve, in some cases, was not connected and was not in the QR software. Again it also should be used to speed up recovery.

2.4 Premature, or too late, change from QR mode to Cool Down and from Transition Mode to Fill Mode slows down the recovery process.

2.5 The transition from Fill Mode to Operate Mode moves the tunnel JT valve(s) too brusquely. The change from 80% open (fill) to ≅57% open (operate) must be done gradually.

3. RECOMMENDATIONS

To improve the performance of the AQR the following changes should be implemented:

3.1 Incorporate control over "all" KV valves except KV5.

3.2 Redefine the values that the corresponding TRQ should reach to close the KV valves. The value possibly will depend on the type of quench.

3.3 Make the change from Transition Mode to Fill Mode based not only on the value of the TR on the end of the string but also on the rate of change of its value.

3.4 Make a smooth closing of the tunnel JT's from 80% to the Maximum Position in the Operate Mode. This can be accomplished by closing the valve 1% every 10 to 14 seconds.

A13 (Cell 1) Software Quench Recovery

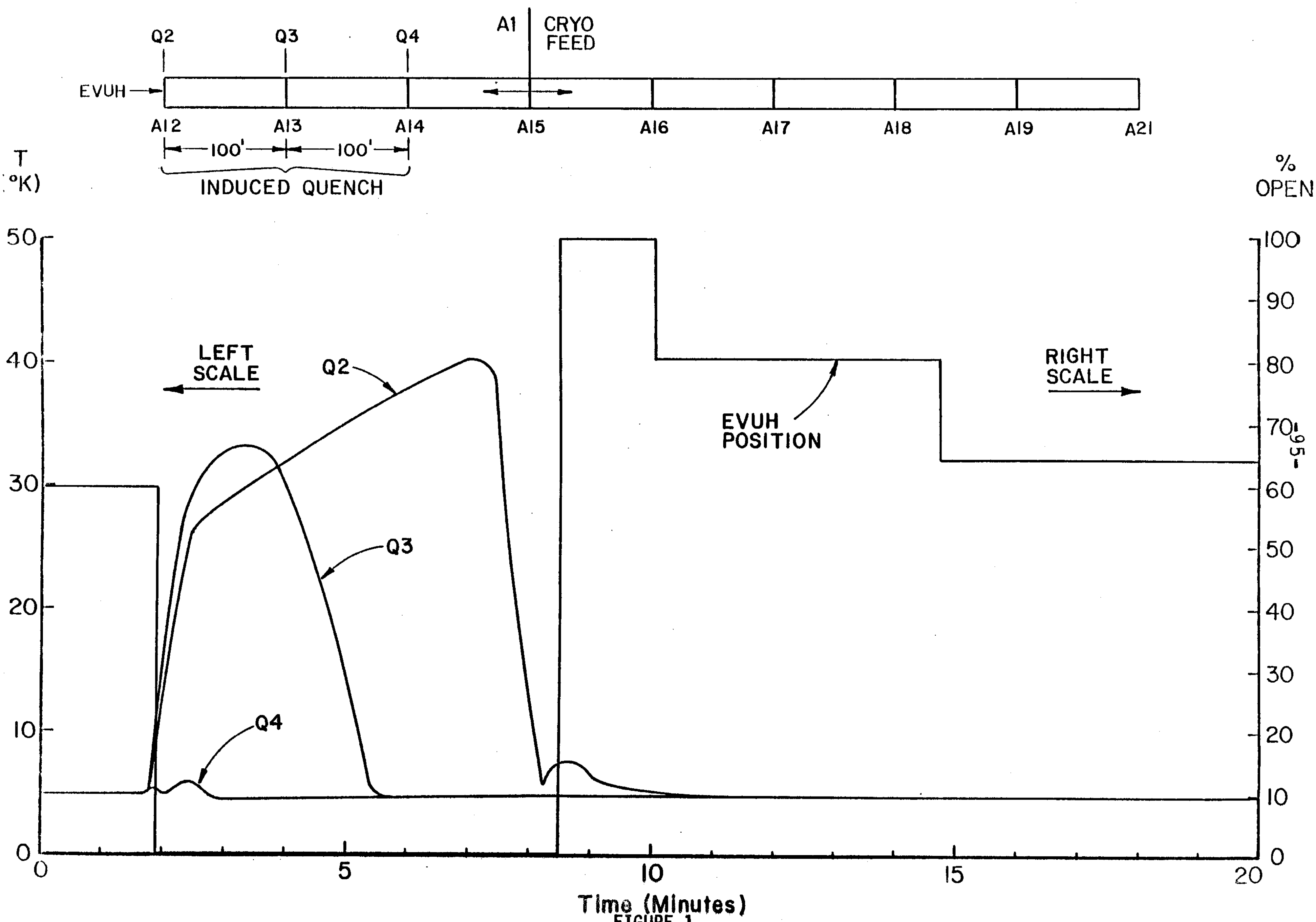


FIGURE 1

A13 (Cell 1) Software Quench Recovery

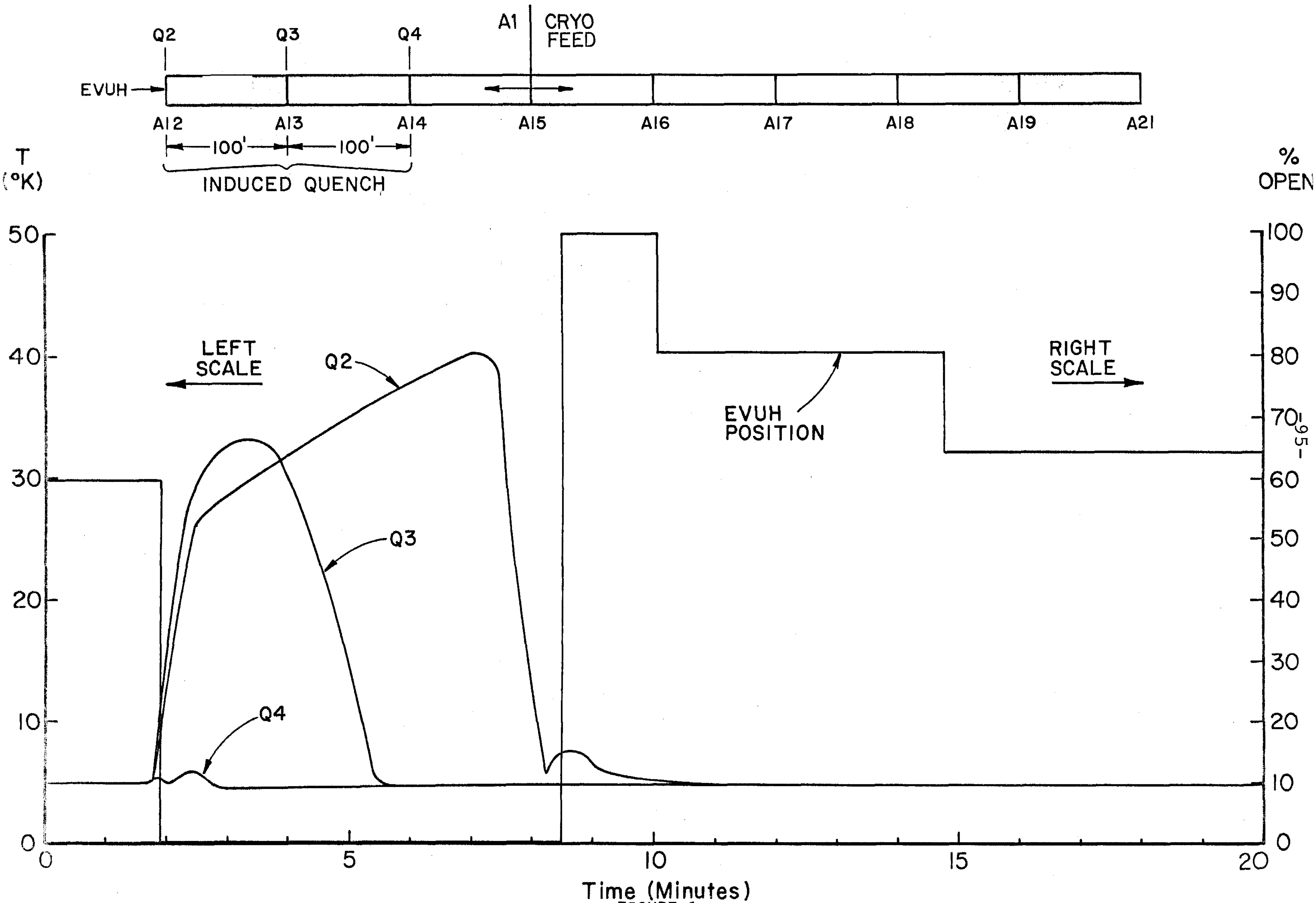


FIGURE 1

A13 (Cell 1) Software Quench Recovery

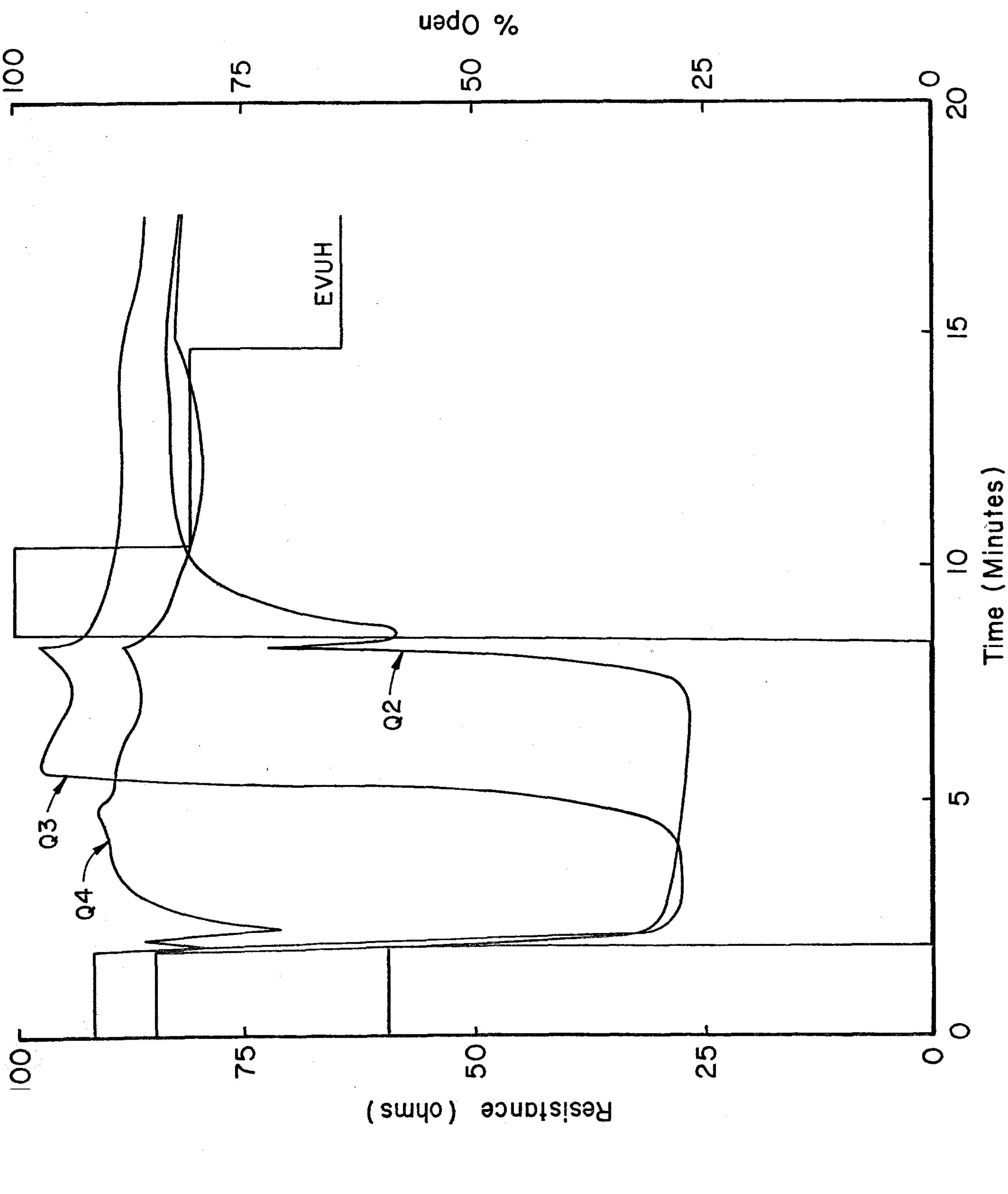
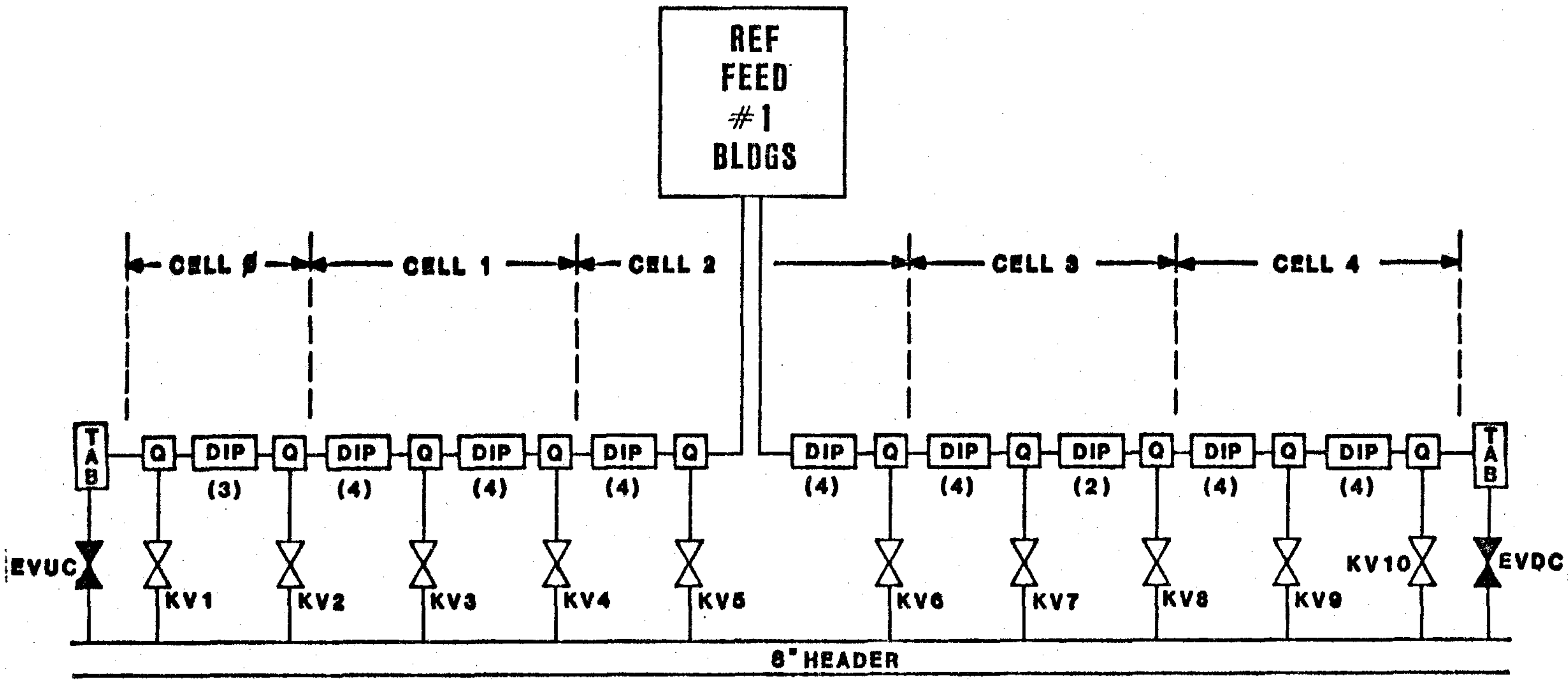


FIGURE 2

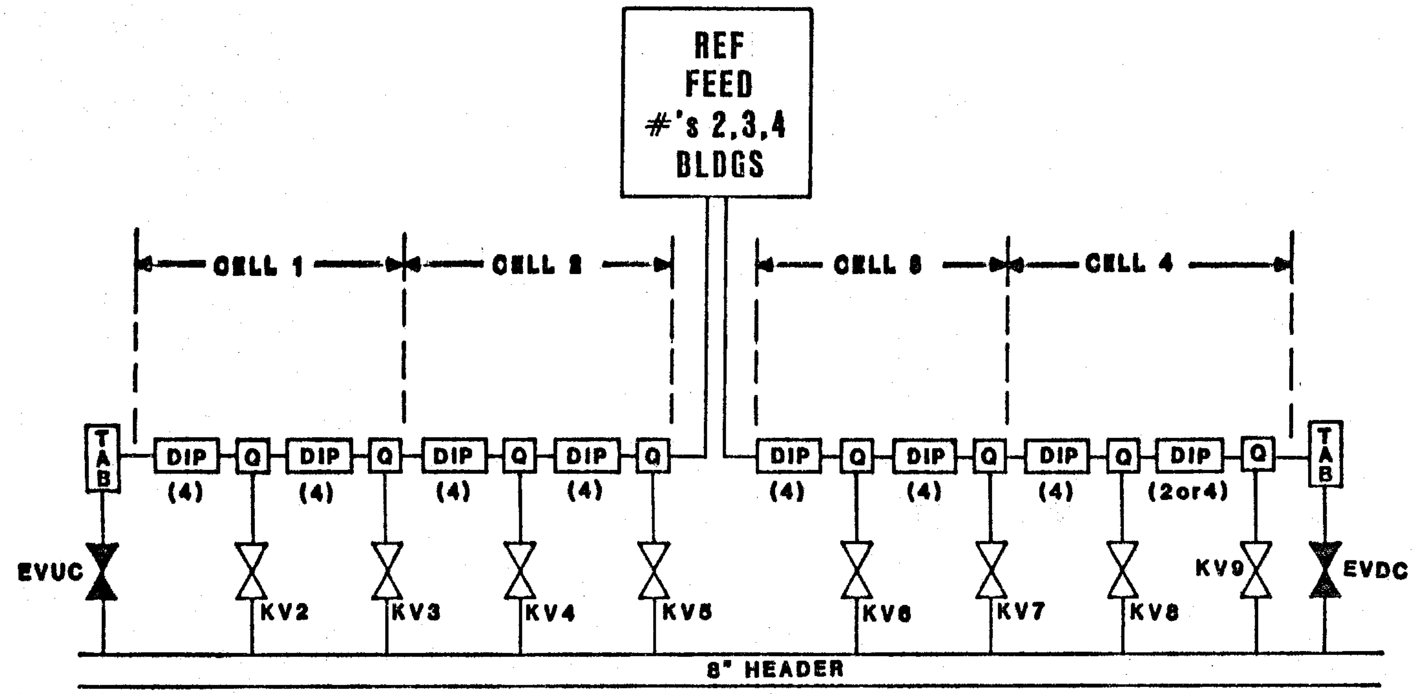
QUENCH RECOVERY VALVES IN #1 BUILDINGS



Q -QUAD
DIP -DIPOLE
KV -IS A SOLENOID CONTROLLABLE KAUTZKY VALVE
EV -IS A NORMAL REFRIGERATOR VALVE

FIGURE 3

QUENCH RECOVERY VALVES
IN #'s 2,3,4 BUILDINGS



Q -QUAD
DIP -DIPOLE
KV -IS A SOLENOID CONTROLLABLE KAUTZKY VALVE
EV -IS A NORMAL REFRIGERATOR VALVE

FIGURE 4

XVI. MAGNET JT AND RELATED CONTROL LOOPS

Manuel I. Martin

0. General

The magnet JT valves have a dual role in the refrigerator system. During Cool Down in the Transition Mode the JTs are 100% open - with the Cool Down valves fully closed - as to quickly divert the flow through the return (2Ø) in the magnets and subcooler into the return side of the heat exchangers of the refrigerator. When the TR in the end of the magnet s string reaches a predefined valûe the corresponding JT valve is closed to 80% and the string enters the Fill Mode.

During Fill Mode the JT is maintained 80% open until the string is full at which time the strings enter the Operate Mode. In Operate Mode the tunnel JT position is controlled by a close loop - through the u-p - that tries to keep the superheat, measured by the corresponding Differential Thermometer (DT), around .1 to .2 psid above the zero of the DT. This technique will assure that in the end of the corresponding magnet string the 20 He will consist of 10% liquid and 90% gas, thus assuring an optimal performance of the cooling of the magnets (see Fig.#1).

The magnets JT values are also used during an automatic quench recovery. In particular, at the end of the automatic quench recovery, when the affected string is put in Transition mode.

Because of the dual role of these valves they have logarithmic bullets (1000:1) and, as a result, it becomes difficult to use them in the Operate Mode when fine adjustments of flow are required.

1. Goals of the A-Sector Refrigerator Tests

During the A-Sector test major goals were defined in relation to the performance of the JT's:

1.1 Find the base line operation of the valves in different conditions of load defined by no ramp of the magnets and different ramping conditions. The base line operation is here defined as the minimum opening of the JT that will assure proper subcooling and that will allow operation of the cryogenic system without interruption.

1.2 Obtain the best possible controlling parameter for the control algorithm.

1.3 Investigate the possibility of obtaining a "single" set of parameters that will allow it to go from Fill Mode to

Operate Mode and from Operate Mode without ramp to different ramping conditions.

2.0 Test Conditions and Results

After several attempts to obtain believable base line measurements it was decided to try to stabilize the control loops of the valves before obtaining the baseline values.

To obtain a set of parameters for the JT loops it was necessary to stabilize the refrigerator plant as well as possible. This was done by assuring that the plant was cool (TR12>80) and that PI14 was as stable as possible with a median value of 13.5 psi. The central liquifier intake valve EVLH was fixed at a proper position to ensure the stability of the refrigerator. The conditions of the different loops of the refrigerator were as shown on Table 1 (refer to Fig.#2).

Table I

Loop #	Input Variable	Set Point	Output Variable	Loop Status	Valve Position
1	PI13	16 PSIG	EVBY	Enable	Normally Closed
2	PI13	14 PSIG	SPWE	Enable	----
3	DT16	*	EVUH	* *	
4	DT19	*	EVDH	* *	
5	PI13	12 PSIG	EVJT	Disable	0 %
6	T17	- - -	SPDE	Disable	Off
7	TI5	NA	EVX1	Enable	----
8	TI5	NA	EVX2	Enable	----
9	TP20	60 Ω	EVLN	Enable	----
10	TI7	----	EVLH	Disable	50 %
11	TP23	23 Ω	EVUN	Enable	----
12	TP24	23 Ω	EVDN	Enable	----
13	TR12	----	EVUC	Disable	0 %
14	TR12	----	EVDC	Disable	0 %

*During the studies of loop stablilty only one of the two (DT16 or DT19) input variables was given a set point equal to its prestablish zero.

**During the stablity studies only the loop under study was enable with the other one disable and its valve set at 50% to eliminate perturbations and crosstalk between loops. The stability studies were done at A3 and A2 starting with loop #3 (DT16/EVUH) at A3 and with the ramp off.

Every time the stability studies were performed the refrigerator system was run for ≈ 2 hours with both tunnel JT loops disable and both EVUH and EVDH 50% open to assure as stable conditions as possible. After this preliminary stabilization of the plant the loop under studies was enable and changes on the following parameters were made - if warranted - every hour.

Loop parameters KO, K1 and ST
Maximum valve position MAXP
Minimum valve position MINP
Maximun value position change MAXO

The first study done with A3EVUH took 16 hours to complete and the studies done on A3EVDH, A2EVUH and A2EVDH averaged 8 hours per loop.

The following parameters were found as the "better compromise".

Table II

Working Parameters for the Tunnel JT Loops

		New	Old
Sampling time	ST	60 seconds	10 seconds
	KO	-8.9	-27
	K1	8.0	25
Maximum position change	MAXO	1%	3%
Maximum error tolerance		∞	.3psid
Maximum valve postion error		0.05	.2

This parameters were defined by May 20, 1982 and used through the rest of the A-Sector test in all three buildings (see apendix A).

After the working parameters for the tunnel JT valves were obtained the other two goals were also achieved.

The base line operations for the different tunnel JT valves were finally establishd as per table III. To obtain these valves a similar tactic as the one used for obtained the working loop parameters was used. Working with one JT valve at the time a large minimum position for the JT valve was chosen and after stable condtions were reached, this valve was decreased 1% every 30 to 45 minutes to achieve stable conditions again - until the corresponding DT became to respond and move fast more than 2 psid above the pre-established zero. Once this value was obtained a minimum position for the JT valve was defined 2% above it for the normal operation of the refrigerator. Table III reflects these "operation minimums" for different ramping conditions.

Table III

Minimum Operating Positions for JT Valves

Ramp	Building	Valve	Minimum Position
3000A	A1	EVUH EVDH	47% 57%
	A2	EVUH EVDH	43% 45%
	A3	EVUH EVDH	46% 47%
3500A	A1	EVUH EVDH	46% 58%
	A2	EVUH EVDH	43% 44%
	A3	EVUH EVDH	48% 48%
4000A	A1	EVUH EVDH	43% 57%
	A2	EVUH EVDH	47% 48%
	A3	EVUH EVDH	52% 55%
No Ramp	A3	EVUH EVDH	41% 47%
	No data for A1 or A2		

With the old ramp parameters for EVUH and EVDH, one scheme to avoid total loss control over the sub-cooling was to maintain a small window between the minimum and maximum position of the JT valves. This window was set at 5%. To achieve the third goal it is necessary to be able to run the loops with a Maximum position of 80% and a Minimum position of 41% to 58% (see Table III). With the new loop parameters it is definitely possible to expand the window between MINP and MAXP to 10% (this was tested 5/30/82) and again in the last 4 Automatic Quench Recoveries when I did not change the MAXP of the JT valves from the 80% required during Fill Mode. Although there is not enough data to categorically say that the third goal was achieved I am confident that it will be possible.

3.0 Observations

During the A-sector test, and in particular during the time specifically spent in the study of the loops, the following conditions were observed.

- A qualitative correlation between loops shown in Table IV (see Fig. #1)

Table IV

	EVBY	SPWE	EVVH	EVDH	EVJT	SPDE	EVX1	EVX2	EVLN	EVLH	EVVN	EVDN
EVBY		S	M	M	S	S	L	L				
SPWE	*		M	M	S	M	L	L				
EVUH				S	M	S	M	M		S		
EVDH					M	S	M	M		S		
EVJT	*	*				L	L	L		L		
SPDE							S	S				
EVx1								S	S	M		
EVx2							*		M	S	L	L
EVLN										L	S	S
EVLH											L	L
EVUN												L
EVDN												

where the upper half shows the correlation between loops defined by:

S→Strong
M→Medium
L→Low
blank→Weak

and the lower half shows the loops that have the same input variable marked with an *.

- With the new parameters for the Magnet JT loops it is possible for the system to have one or both DT (superheat) to go above the maximum limit of the device and not lose supercooling in the magnets as long as the loops bring the DTs below 1 psid within 7 to 8 minutes (See Fig. #3).

- Oscillations of one (or both) DT between the zero (minimum value of the device) and 2.75 psid without noticeable effect in the cryogenic system (see Fig. #4).

- There is a strong coupling between the DT's,but it seems to be an indirect coupling through TI7 and the wet engine output, more than a direct crosstalk.

- During the stabilization process of the Magnet JT Loops a complete PID algorithm - with derivative gain - was tested without any appreciable improvement. High derivative gain (GD) made the loop unstable (see appendix A).

- To be able to have good control over the cryogenic system the movements of the Magnet JT Valves must be smooth (increments below 1%) otherwise a strong perturbation in the flow of helium through the magnets creates a situation where the DTs increase and subcooling is lost.

  It is important to recall that a change of 1% in the JT valve position corresponds to a 10% change of flow through the magnets.

- Because of the above a strong correlation exists between the sampling interval (ST) and the maximum valve movement (MAXO) allowed in the Magnet JT Loops. Because of this the "fast loop control", that generates corrections every 2 seconds instead of every ST seconds, that takes effect when the error between desired value of the of the input variable and its measured value exceeds the Maximum Error Tolerance (MET) cannot be used. This is accomplished by giving a very high value to MET.

4.0 Recommendations

Based upon the tests performed and the results obtained the following recommendations are made:

- Provide for finer adjustment of the Magnet JT valves. This will require possible modification in the valve actuator card and/or in the actuator itself. A minimum repetitive change of .25% is a good goal.

- Investigate further the possibility of using three constants corrective algotirhm (a PID).

- Investigate the possibility of using a different variable to control the position of the valves. It could be a "quality meter" located between the magnet string and the subcooler as G. Mulholland suggested, or a derived quantity that will give a good measurement of the percentage of gas/liquid that reaches the subcooler in the return side (2Ø).

- Study the possibility to muffler the output of the wet engine so PI14 does not see changes in pressure due to the engine cycle, especially at low speeds.

- Study further the correlation between EVLH (SPDE in stand alone mode) loop and the Magnet JT Loops.

- Obtain better control of both input pressure (3" high pressure) and output pressure (8" low pressure) because changes are then reflected as oscillations on the DTs. In particular, variations of pressure in the 8" header create unstable conditions.

Appendix A

Closed Loop Control Algorithm

The expression used by the μ-processor is

$$P(n) = 0(n) + P(n-1) \qquad A.1$$

with

$$0(n) = K_o E(n) + K_1 * E(n-1) \qquad A.2$$

where

$P(m)$ = valve position at time $m*ST$
$0(m)$ = change of valve position at time $m*ST$
$E(m)$ = Error between the desired value and the measured value of the input (controlled) variable.
ST = Sampling and correcting interval measured in seconds

In this case it is easy to give physical meaning to K_0 and K_1.

By combining A.1 and A.2 we have $P(n) = K_0*E(n) + K_1*E(n-1) + P(n-1)$ A.3

that can be expanded to

$$P(n) = K_0*\left[E(n) + E(n-1)\right] + K_1*\left[E(n-1) + E(n-2)\right] + P(n-2) \qquad A.4$$

and recursively

$$P(n) = K_0*\sum_{j=1}^{n} E(j) + K_1 \sum_{j=0}^{n-1} E(j) + P(0) \qquad A.5$$

If we assume that $E(0) = 0$ adding and subtracting $K_1*E(n)$ we can express A.5 in the following way

$$P(n) = -K_1*E(n) + (K_0+K_1)\sum_{j=0}^{n} E(j) + P(0) \qquad A.6$$

and multiplying and dividing the second term on the right hand side of A.6 by ST we have

$$P(n) = -K_1*E(n) + \frac{K_0+K_1}{ST}*\sum_{j=0}^{n} E(j)*ST + P(0) \qquad A.7$$

This expression can be written as follows

$$P(n) = PG*E(n) + IG*\sum_{j=0}^{n} E(j)*ST + P(0) \qquad A.8$$

where

PG = Positional Gain
IG = Integral Gain
$P(0)$ = Constant equal to the starting position of the valve.

Comparing A.7 and A.8 we can write directly the following relationship between PG, IG, K_0 and K_1.

$$PG = -K_1 \qquad A.9$$

$$IG = (K_0+K_1)/ST \qquad A.10$$

Or

$$K_0 = IG*ST + PG \qquad A.11$$

$$K_1 = -PG \qquad A.12$$

So the correcting algorithm (A.2) used with only two constants K_0 and K_1 is equivalent to a PI algorithm. To obtain derivative gain (to generate a PID algorithm) it is sufficient to add one more term to A.2 so it becomes

$$O(n) = K_0*E(n) + K_1*E(n-1) + K_2*E(n-2) \qquad A.13$$

and A.5 becomes

$$P(n) = K_0*\sum_{j=2}^{n} E(j) + K_1*\sum_{j=1}^{n-1} E(j) + K_2*\sum_{j=0}^{n-2} E(j) + P(0) \qquad A.14$$

and, by the same reasoning as before, adding and subtracting

$K_1*E(n) + K_2*\left[E(n) + E(n-1)\right]$ and with $E(0) = E(1) = 0$

we obtain

$$P(n) = -(K_1+2K_2)*E(n) + (K_0+K_1+K_2)*\sum_{j=0}^{n} E(j) +$$

$$K_2*\left[E(n) - E(n-1)\right] + P(0) \qquad A.15$$

or

$$P(n) = PG*E(n) + IG*\sum_{j=0}^{n} E(j)*ST + DG*\frac{E(n)-E(n-1)}{ST} + P(0) \qquad A.16$$

from where we can obtain the following relationship between PG, IG, DG, K_0, K_1 and K_2

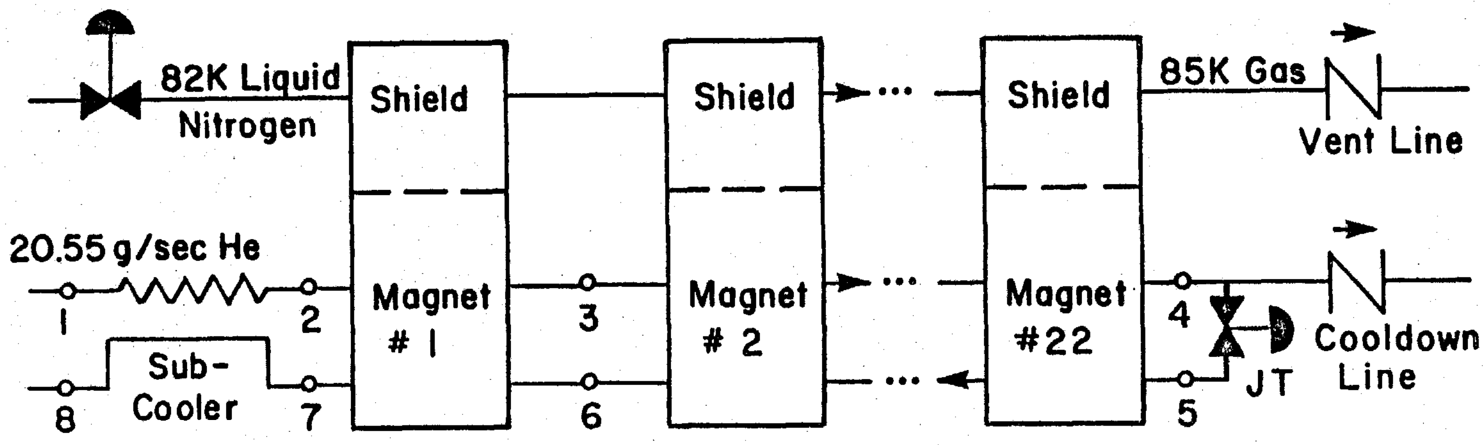
$$PG = -(K_1 + 2\,K_2) \qquad A.17$$
$$IG = (K_0 + K_1 + K_2)/ST \qquad A.18$$
$$DG = K_2*ST \qquad A.20$$

or

$$K_0 = PG + IG*ST + DG/ST \qquad A.21$$
$$K_1 = -PG - 2*DG/ST \qquad A.22$$
$$K_2 = DG/ST \qquad A.23$$

where DG is equivalent to a differential gain.

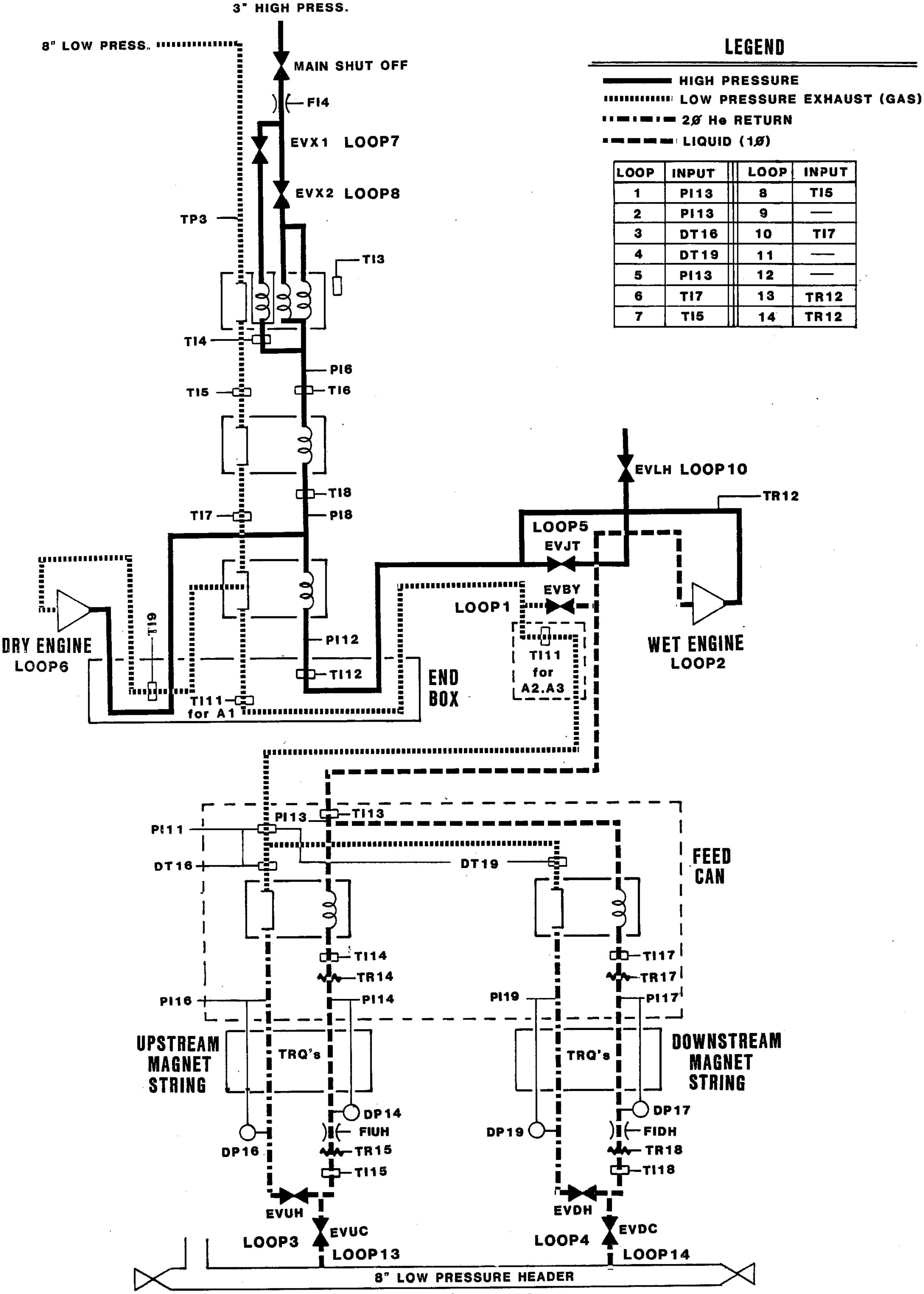
In this case the correcting algorithm, using three constants, is equivalent to a typical PID algorithm.



POINT	T(K)	P_{atm}	$H_{J/g}$	% LIQUID
1	4.90	1.8	14.22	100.
2	4.50	1.8	11.20	100.
3	4.55	1.8	11.47	100.
4	4.60	1.8	11.75	100.
5	4.47	1.25	11.75	96.
6	4.42	1.2	27.49	13.
7	4.42	1.2	27.99	10.
8	4.52	1.2	31.01	0.1K Super Heat

Figure 1

AS IT FLOWS THROUGH A REFRIGERATOR BUILDING AND ASSOCIATED MAGNET STRINGS.



LOOP	INPUT	LOOP	INPUT
1	PI13	8	TI5
2	PI13	9	—
3	DT16	10	TI7
4	DT19	11	—
5	PI13	12	—
6	TI7	13	TR12
7	TI5	14	TR12

Figure 2

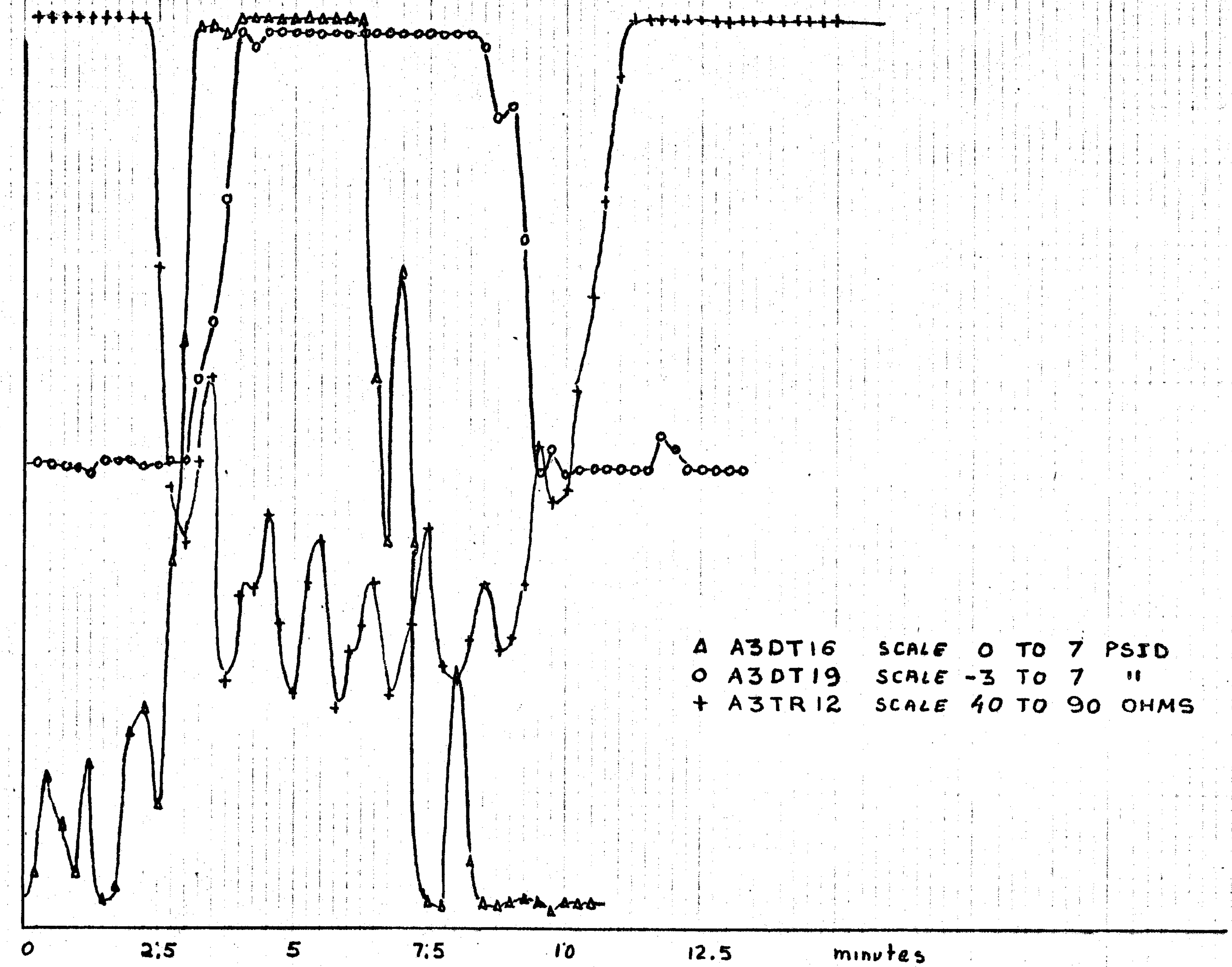
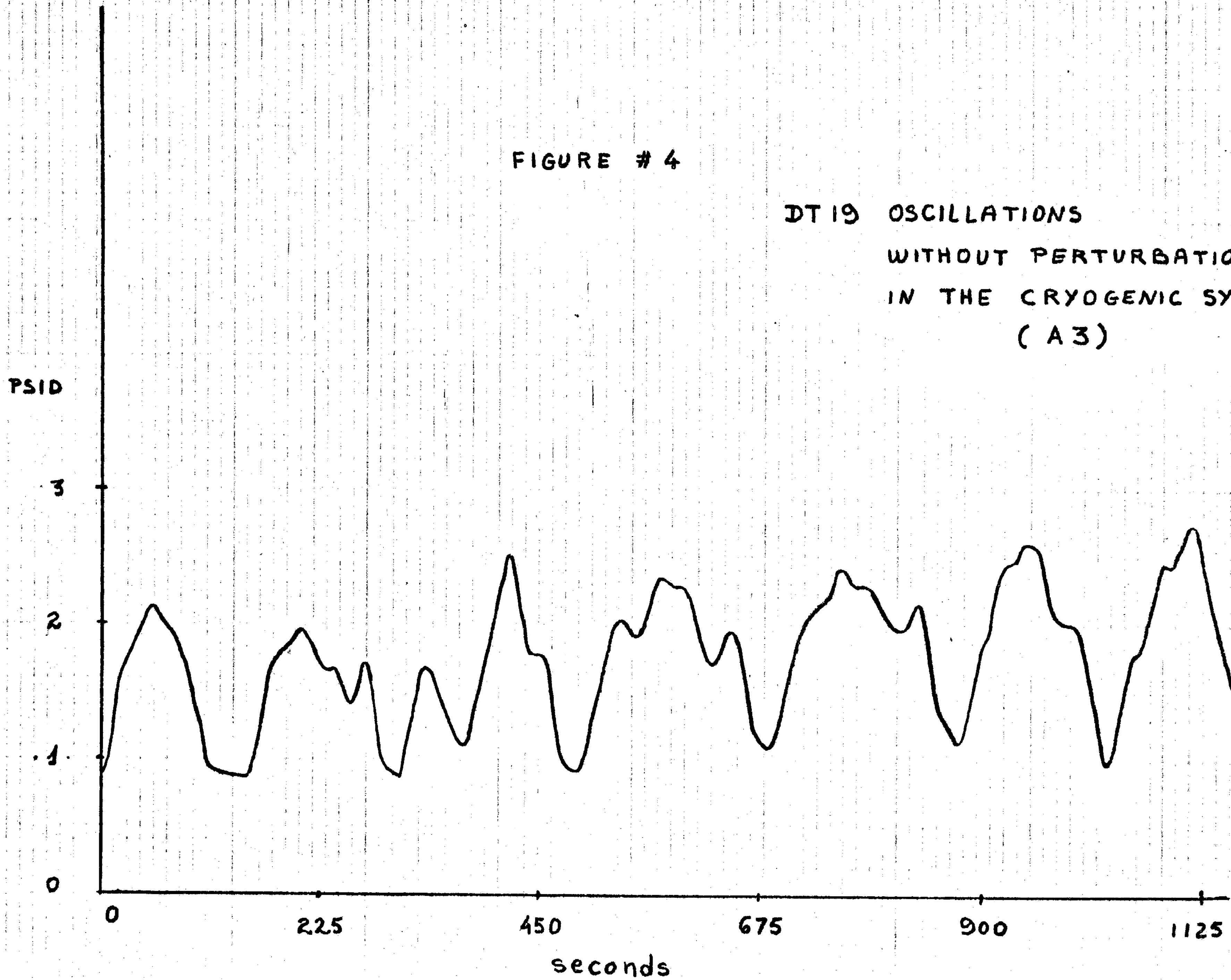

Δ A3DT16 SCALE 0 TO 7 PSID
O A3DT19 SCALE -3 TO 7 "
+ A3TR12 SCALE 40 TO 90 OHMS
0
2.5
5
7.5
10
12.5
minutes

FIGURE # 3

FIGURE # 4

DT 19 OSCILLATIONS
WITHOUT PERTURBATIONS
IN THE CRYOGENIC SYSTEM
(A3)



Fermilab

XVII. SIMULATOR COMPARISONS

Henry R. Barton, Jr.

INTRODUCTION

This report summarizes simulation work in two areas. The first area is the cryogènic behavior of the magnet cryoloop with and without closed loop control of the JT-valve applied. This investigation shows that control of the magnet string imposes a preferred control algorithm. The second area of study is an attempt to evaluate the currently implemented control algorithm. Simulation studies show that the current algorithm does not behave as a PID feedback system.

CRYOLOOP SIMULATION

The existing simulator is capable of predicting the cryogenic operating behavior of the magnet system under upset conditions. It is recommended that the heat input to the cryoloop be calculated in a feedforward algorithm and that this calculation be used with supervisory control of the JT-valve position to modify the setpoint so as to produce a helium flow which is consistent with the magnet heat load. To the extent that this feedforward strategy is not precisely accurate in determining the helium flow required to exactly match the magnet heat load, a feedback loop will be required to maintain the delicate balance between flow and heat load.

The first simulation study performed involved a small (10%) increase in flow through the magnet string without any change in heat load. The effect of this increase in refrigeration is shown in Figures 1 and 2. These results can be characterized as a smooth change in cryogenic conditions during a slow (~20 minutes)transition from one steady state condition to another steady state condition. No control mechanism was in operation during this first simulation so that the nature, the duration and the magnitude of the uncorrected effect could be determined.

The results of this first simulation suggest the appropriate control scheme. Since an uncorrected error in helium flow produces a steady state condition, a predominant component of the control algorithm must be reset action, which is a feedback signal proportional to the time integral of the error signal. This control algorithm was incorporated in the simulator to correct the helium flow. The simulation results using closed-loop control of the flow are shown in Figures 3 and 4. The magnitude of the effect has been reduced and the transient is fully corrected in 30 minutes.

TEVATRON CONTROL ALGORITHM

In order to study the existing JT control system, the control algorithm currently used was incorporated in a simulation program. Preliminary studies consist of giving the current system a well defined error signal ε and observing the resulting change in JT-valve position produced by the control algorithm.

Figures 5 and 6 show that for specially chosen values of the loop tuning constants, the algorithm is purely proportional without any derivative action. Figures 7 and 8 show that proportional plus integral action is possible if the magnitudes of the K_0 and K_1 differ; however, no derivative action is produced.

RECOMMENDATIONS

1. Moving the JT-valve produces an effect which requires 20 minutes to develop; therefore, no attempt should be made to control transients having a period in the range 1 second to 1 minute.

2. All the commercial, computer control systems retain the concept of the PID control algorithm and there is no indication that this terminology is outdated. The use of proportional, integral and derivative gains to tune the control loops is well understood and convenient.

3. Any proposed control scheme can be given a preliminary test on the simulator. There is no indication that a control algorithm which fails during simulations would function satisfactorily on the Tevatron.

4. Quality meters could be installed in the spool pieces and the output of these instruments would be a particularly good input to the control loop.

5. The appropriate sampling period for control loop input is an area for continued study. The use of rapidly sampled data in a filtering scheme is under consideration.

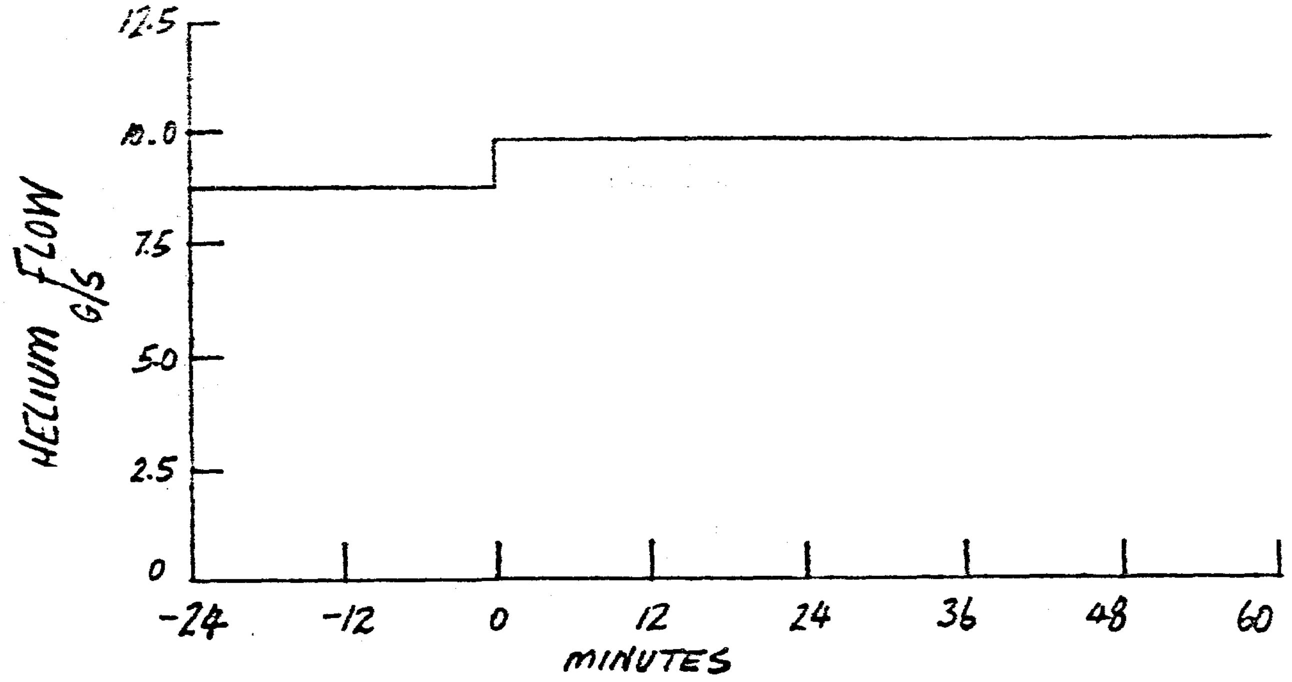


Figure 1.

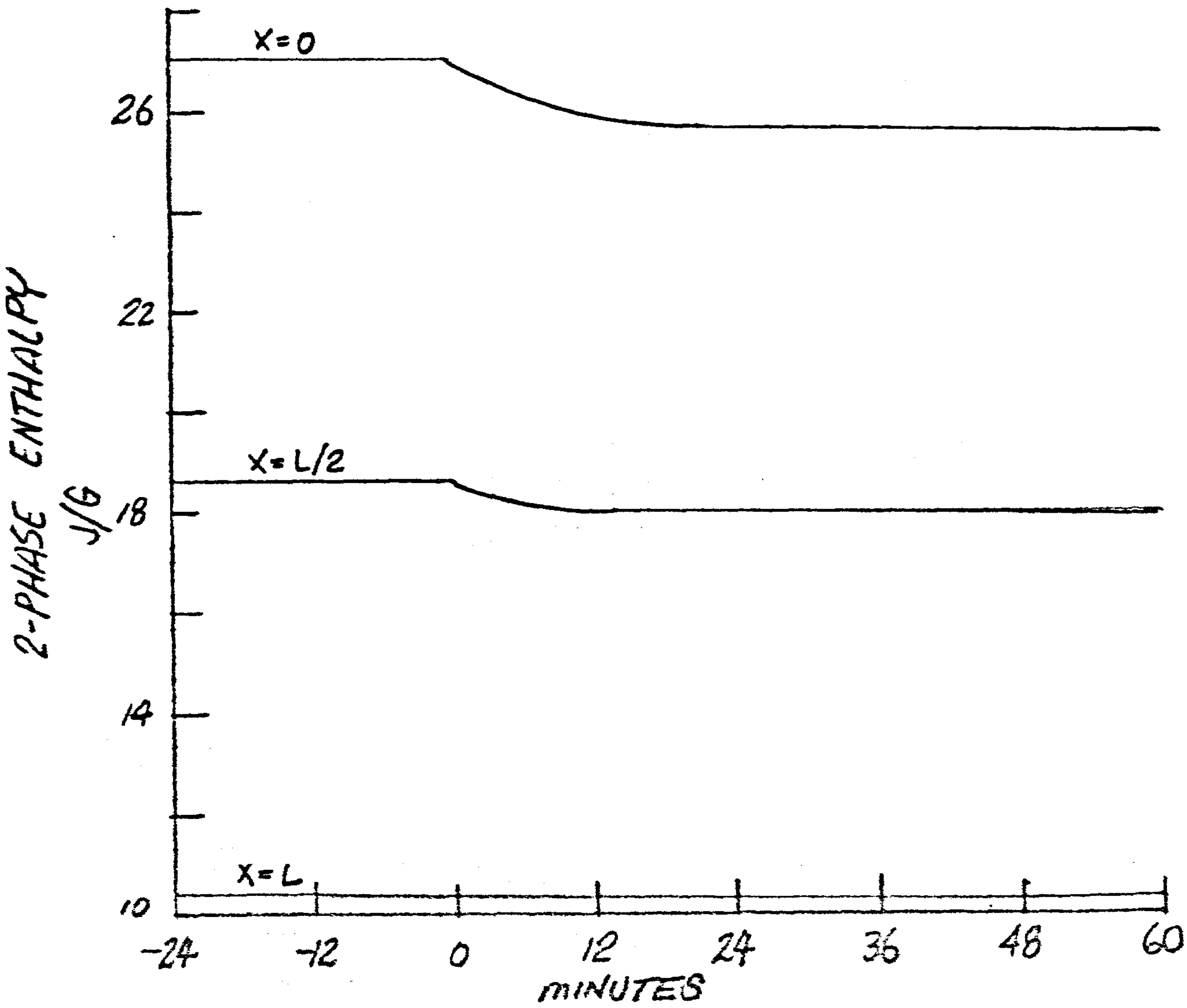


Figure 2.

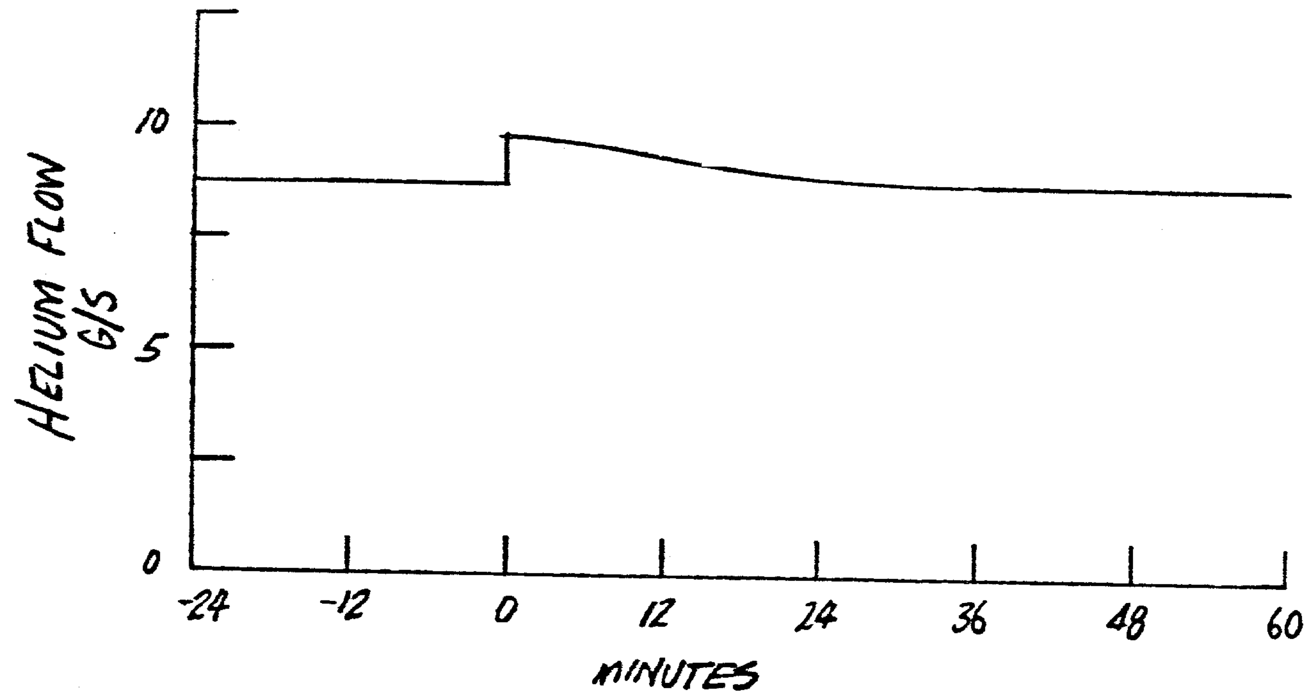


Figure 3.

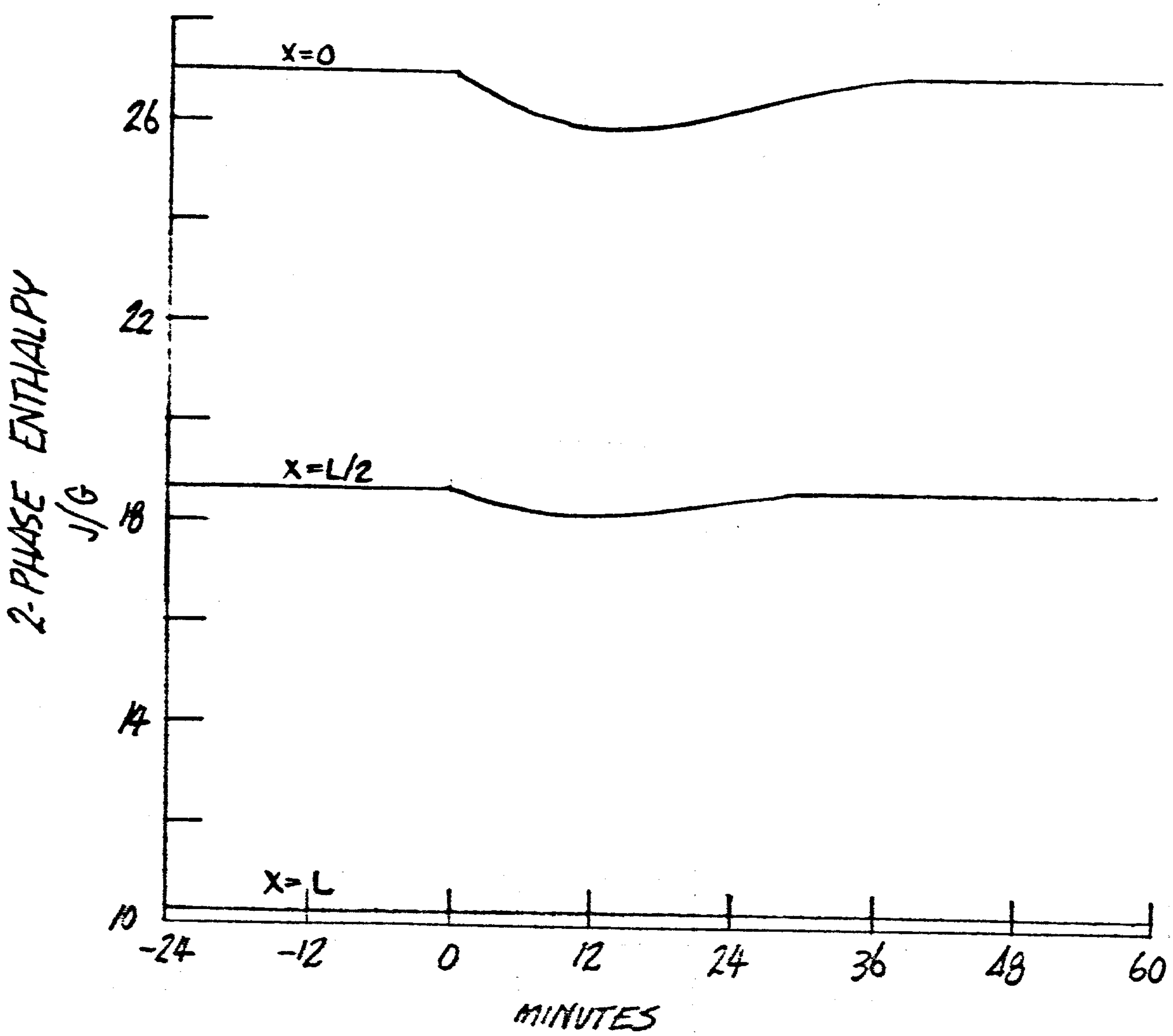


Figure 4.

$K_0 = 1.1 \qquad K_1 = -1.1$

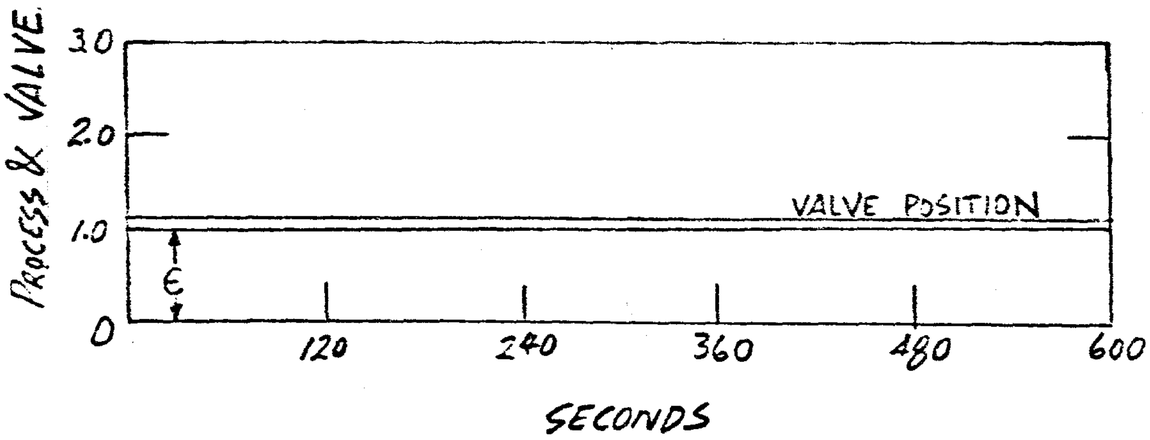


Figure 5.

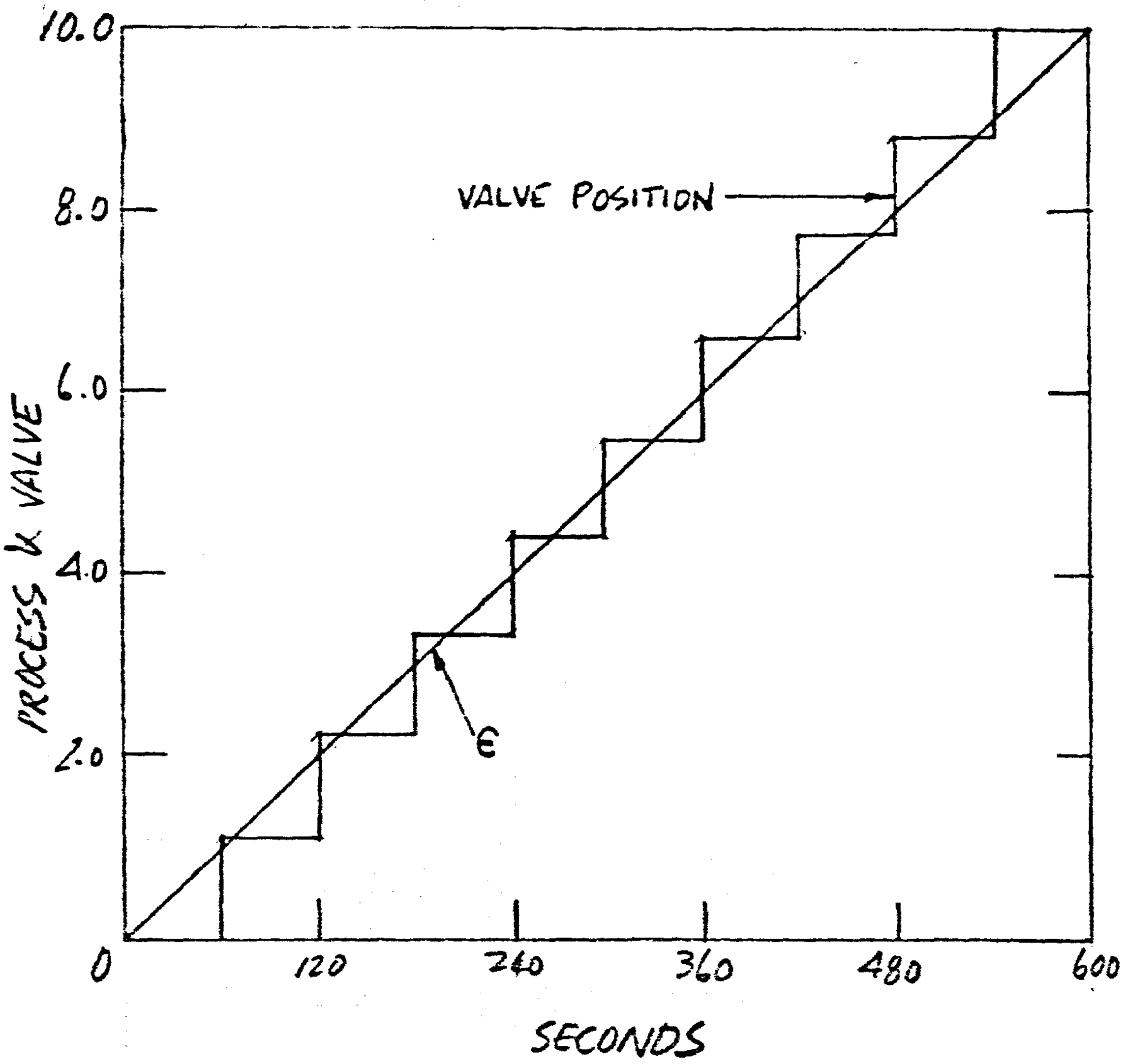


Figure 6.

$K_0 = 1.1 \qquad K_1 = -1.0$

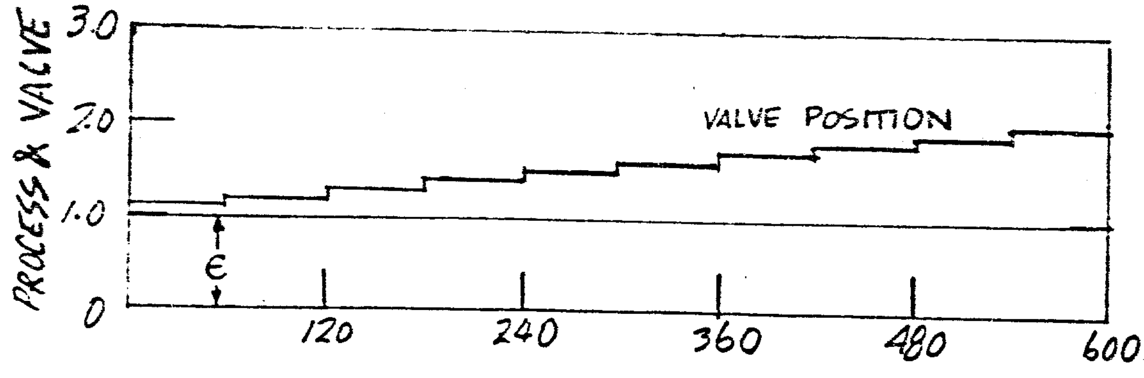


Figure 7.

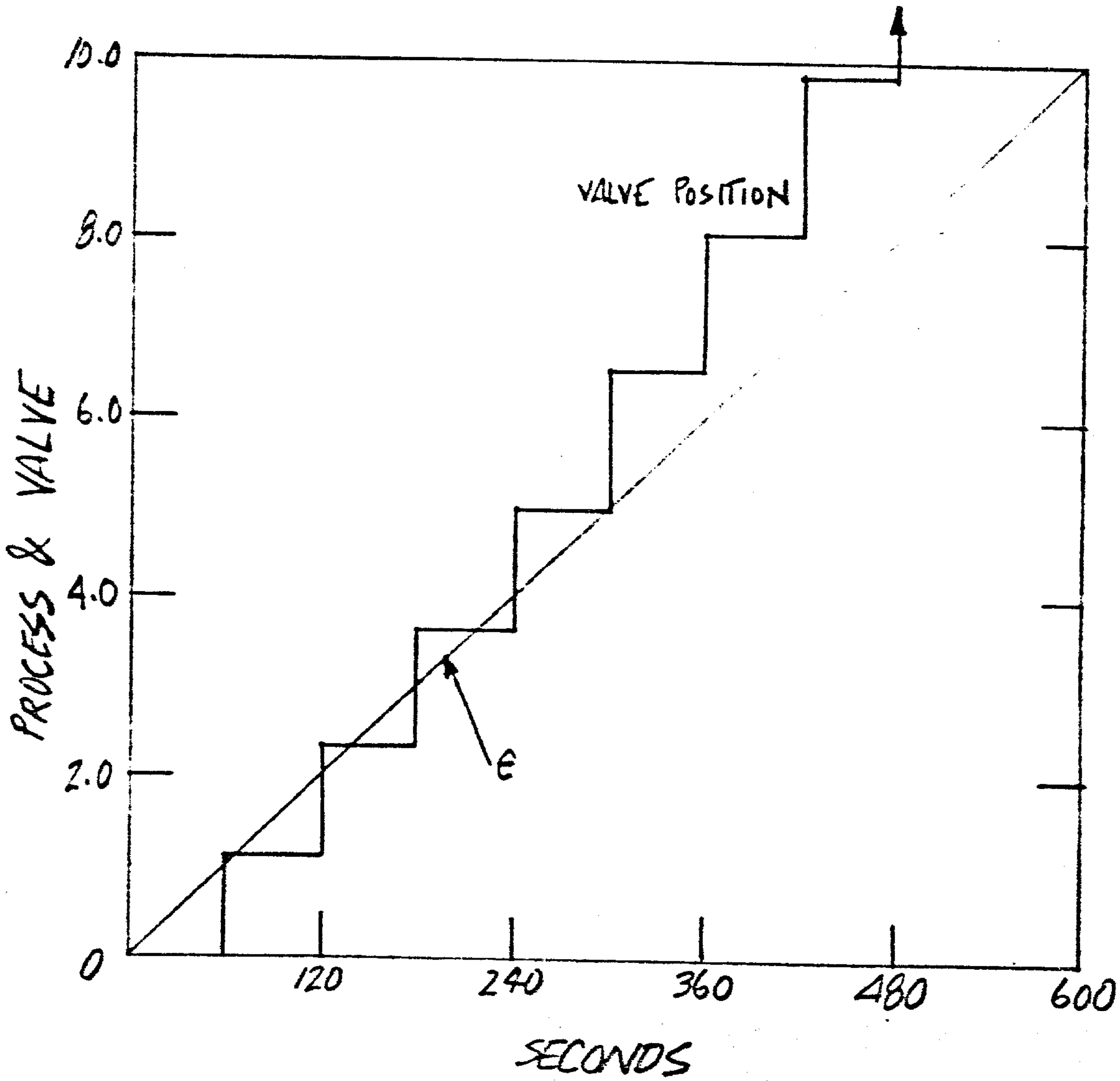


Figure 8.

XVIII. REFRIGERATOR FAILURES

J. Savignano/J. C. Theilacker

During the recent A-Sector testing, several refrigerator problem/failures required attention. Recognizing the problems and intiation of a fix is an important part of a system startup. Several groups are working toward solutions to the following A-Sector problems in order to achieve smoother future operation.

Several refrigerator based problems have been addressed in more detail in other sections. These include: expansion engine efficiency rolloff/failures - Section VI, Kautzky relief valve failures - Section XIV, power lead leaks - Section X, and transfer line oscillations - Section IV. The transfer line oscillation early in the run reminded us that the A1½ and A2½ expansion cans had old design relief valves which relieve at lower pressure. These reliefs were updated to the standard relief used throughout the rest of the ring. This prevented mild pressure oscillations from allowing LN_2 or LHe reliefs from weeping.

Other problems Included:

High Pressure Valve Leakage

During the run, we noticed high pressure valve stem packings leaking to atmosphere. One of the valves is the main refrigerator shutoff valve (MV101H), a Cooper ball valve. After several cycles on the actuator, the stem packing loosened.

The two high pressure helium supply valves to heat exchanger #1 and 2 (EVX1 and EVX2) also leaked in the stem packing. These Valtek valves are flow control valves with electric actuators. There is a fine line between curing the leak and stalling the actuator when tightening the packing. Higher torque actuators will be used in these location.

8" Header Relief Valves

Several problems arose concerning the 8" header quench relief valves. The first major quench blew snow covers off which were held down by a ½" bolt. A new snow cover is in design.

During A-Sector quenching, refrigerator building hatch covers were blowing off the building. This is due to the high velocity (low pressure) relief flow drawing the hatch up into the flow stream. Latches are being added to the hatches to hold them down, but still allow them to be removed for transfer line U-tube repair.

Late in the run, we developed a problem with the A-3 8" relief valve. During a quench, the O-ring seal blew out. Upon inspection, the problem was found to be an oversized V-groove. A new relief flange will be welded on this summer.

Control Valve Stems

The single, potentially most severe problem encountered during the run involved bellows failure on the valve stems controlling liquid helium/nitrogen removal from the CHL transfer line. Serious insofar as the job of replacing the bellows involved valve stem removals while liquid was present in the valve body. In the case of the liquid helium, only cold, dense gas could exit the valve housing upon stem removal. However, the removal of the liquid nitrogen valve stem poses another problem insofar as liquid can splash out of the valve body upon stem removal. The only sure way to prevent problems is to pull the "U" tube from the transfer line and isolate the liquid helium/nitrogen from the building which entails shutting down a magnet string, representing ring down time of some duration.

As the weather turned more humid, the valve stems condensed moisture and the moisture ran down the stems into the bellows area and froze. Actuating the valves--which is a continual process during normal operations--caused the frozen bellows to rupture. Escaping cryogenic gas froze out the "O" ring seal and vented into the refrigerator buildings. This was not an isolated happening as all three of the helium valves bellows failed--one failed twice. A test showed an actual temperature of $-50^{o}C$ present in the bellows area during refrigerator operations; any moisture here poses severe problems. To fix to this problem we are adding piston rings to the valve to prevent oscillations which lower the bellows temperature. In addition we are installing a lip seal around the valve stem to prevent moisture access into the bellows area. See Fig. 1. At present, new units are under construction to replace those valves and eliminate this problem.

Instrumentation

Most instrumentation problems involved improperly charged vapor pressure thermometers. Although all vpt bulbs were helium mass spectrometer checked, two failed when cooled in operation and lost their charge. As these were internal failures inside the outer vacuum shields, no repair was possible. During the summer shutdown they will be repaired. Another vapor bulb had a solder plug in the charging line -- at the tunnel connection -- cutting the plug out and recharging solved this problem. Because of small inconsistencies in the readouts, the purity of the charging gas was suspect. Although certified 99.999% pure, gas was used in charging the vpt's; perhaps connecting the cylinders to the units introduced contaminates into the gas. New, evacuatable regulators with check valves are now used to connect the gas cylinders to the units and a new procedure for vpt charging has been written to circumvent this problem.

Conclusion

The efforts of many groups enabled us to overcome the problems which arose allowing a successful A-Sector run. Continued efforts throughout this summer and fall will help us achieve a smooth operation of the half-ring and eventual full-ring.

FIGURE 1

BELLOWS VALVE
AS
BUILT

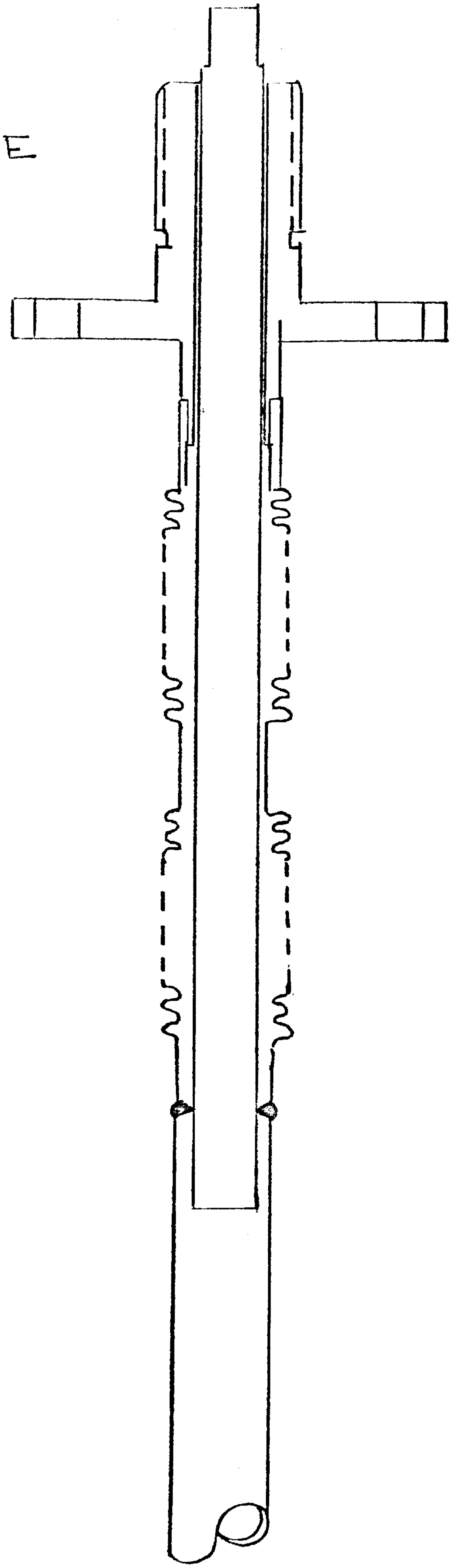


FIGURE 1.A